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Pressure Vessel and System Design

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18.2

Pressure Vessel and System Design*

Contents

1.0 Introduction 1
2.0 Hazards of Pressure Vessels and Systems 2
3.0 Documentation for Pressure Vessels and Systems 2
3.1 Plant Engineering (Livermore) (PEL) Standards 2
3.2 Equipment Requiring an Engineering Safety Note 3
4.0 Design Controls for Pressure Vessels 3
4.1 Design Criteria 3
4.2 Material Selection 5
4.3 Design Considerations 6
4.4 Calculation Guide for Ductile Vessels 8
4.5 Stored Energy 13
4.6 Testing 13
4.7 Retesting and Inspection 14
4.8 Expansion and Compression of Gases 15
4.9 SF-3 Pressure Vessels 15
4.9.1 Materials 17
4.9.2 Tensile Testing 17
4.9.3 Toughness Testing 17
4.9.4 Compatibility 18
4.9.5 Welded Vessels 18
4.9.6 Nondestructive Testing 18
4.9.7 Pressure Testing and Labeling 18
4.10 War Reserve Vessels 19
4.10.1 Documentation 19
4.10.2 Design Criteria 19
4.10.3 Handling 19
5.0 Design Controls Gas-Pressure Containment Vessels 19
5.1 Special Shipping Requirements 20
5.2 Design Safety Factors 20
5.3 General Design Requirements 20
5.4 Testing and Labeling 21
6.0 Design Controls for Pressure Systems 22
6.1 Precautions 22

* Minor revision

6.2 Pipe and Tubing 22
6.3 Pipe and Tube Support 23
6.4 Fittings 24
6.4.1 National Pipe Taper Thread (NPT) Fittings 24
6.4.2 Straight-Thread (Face Seal) Fittings 24
6.4.3 Flare Fittings 25
6.4.4 Flareless or Bite-Type Fittings 26
6.4.5 Coned and Threaded Connections 28
6.5 Valves 30
6.6 Relief Devices 33
6.7 Pressure Gauges 34
6.8 Flexible Hose 35
6.9 Flash Arrestors and Check Valves 36
6.10 Regulators 36
6.11 Manifolds 37
6.11.1 Industrial Gas Cylinder Manifolds 37
6.11.2 Safety Manifolds 38
6.12 Temperature Considerations 38
6.13 Installing Pressure Systems 39
6.14 Pressure Testing 40
7.0 Work Standards 40
7.1 Work Smart Standards 40
7.2 Other Required Standards 41
8.0 References 41
9.0 Resources for More Information 42
9.1 Contacts 42
9.2 Lessons Learned 42
9.3 Other Sources 42

Appendices

Appendix A Terms and Definitions 44
Appendix B Example of an Engineering Safety Note 46
Appendix C ASME Pressure Vessel Code Guide 66
Appendix D Piping and Tubing Pressure Ratings 67
Appendix E Metric Guide 85
Appendix F Joint Efficiencies 100
Appendix G High-Pressure Drawing Symbols 102

Tables

Table 1.	Specifications of materials and alloys for use in pressure vessel.	7
Table 2.	Suggested pipe support spacing (Ref. 13).	23

Figures

Figure 1.	LLNL documentation requirements for pressure equipment.	4
Figure 2.	Relationships between test pressures, the MAWP, and MOP.	9
Figure 3.	Some acceptable types of unstayed flat heads and covers.	10
Figure 4.	LLNL pressure tested label for manned-area operation.	15
Figure 5.	LLNL pressure tested label for remote operation only.	15
Figure 6.	Remote operation label (silver on red).	15
Figure 7.	Amagat chart.	16
Figure 8.	Typical 1/16-in. high-pressure compression fitting.	28
Figure 9.	Typical 1/4-in. high-pressure coned and threaded fitting (60,000 psig).	29
Figure 10.	Typical high-pressure coned and threaded connection (to 60,000 psig).	29
Figure 11.	Type 1 high-pressure valve.	31
Figure 12.	High-pressure valve body patterns.	32
Figure 13.	Safety manifold system.	39

18.2

Pressure Vessel and System Design

1.0 Introduction

This document contains requirements for all pressure vessels and systems used at LLNL. Appendix A contains terms and definitions, and Appendices B through G contain supporting information. Pressure designers and experimenters shall fully understand these requirements or ask a pressure consultant or the pressure safety manager for assistance. Document 18.1, "Pressure," in the *Environment, Safety, and Health (ES&H) Manual* specifies training requirements and responsibilities of individuals who work with pressure vessels and systems. All workers and organizations shall refer to Document 2.1, "Laboratory and ES&H Policies, General Worker Responsibilities, and Integrated Safety Management," in the *ES&H Manual* for a list of general responsibilities. The requirements in Section 3.0 of this document do not apply to the systems listed below.

- Unmodified compressed gas or liquid cylinders approved by the Department of Transportation (DOT) and the appropriate regulators.
- Utility systems that
  - — Comply with "Laboratory Gas Systems," PEL-M-13200. This standard can be found in "LLNL Facility Design Standards."—Operate at a maximum allowable working pressure (MAWP) of no more than 2 MPa gauge (300 psig).
  - — Are inspected at installation and subsequently maintained by the Plant Engineering Department.
- Refrigeration systems that comply with the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code[1] and applicable Air Conditioning and Refrigeration Institute (ARI) standards.
- Systems that operate at an MAWP less than atmospheric pressure. (Design guidance for these systems is given in the ASME Boiler and Pressure Vessel Code[1] and Mechanical Engineering *Design Safety Standards*, M-012, Section 3.3, "Vacuum Systems." However, the requirements apply to vacuum systems that may be pressurized (i.e., for purging or backfilling).
- ASME-coded air-pressure tanks, liquefied petroleum gas tanks, anhydrous-ammonia tanks, and fired steam boilers that are inspected periodically in accordance with 8 CCR §§ 450–560, "Unfired Pressure Vessel Safety Orders"

or ASME Boiler and Pressure Vessel Code.[1] The responsible designer shall notify management whenever such systems are installed.

The LLNL Pressure Safety Program is administered and monitored by the Hazards Control Department through the pressure safety manager. Responsible Individuals shall conduct all pressure work safely in accordance with the *ES&H Manual* with particular attention to Part 18. The pressure safety manager oversees the work of pressure installers and pressure inspectors and coordinates all pressure safety training. The Mechanical Engineering Department Safety Committee and the pressure safety manager provide technical guidance for the program. In addition, pressure consultants are available to help on pressure safety design and to answer engineering questions.

2.0 Hazards of Pressure Vessels and Systems

The hazards presented to equipment, facilities, personnel, the public, or the environment because of inadequately designed or improperly operated pressure vessels and system include blast effects, shrapnel, fluid jets, release of toxic or asphyxiant materials, contamination, equipment damage, personnel injury, and death.

3.0 Documentation for Pressure Vessels and Systems

Designers shall maintain a design file for all pressure vessels and systems in accordance with requirements specified by their department. This file shall contain appropriate records (e.g., Engineering Safety Note, if required) that shall be reviewed at least every three years when vessels or systems are inspected. Revisions and addendum to the file shall be made as appropriate.

3.1 Plant Engineering (Livermore) (PEL) Standards

Normal pressure systems designed or fabricated by Plant Engineering (PE) Livermore personnel or by contract employees working through a PE field team shall conform to the requirements in “Laboratory Gas Systems,” PEL-M-13200. Any deviation from this standard requires approval by an LLNL pressure consultant. Systems that are not covered by PEL-M-13200 require an ESN or equivalent documentation (e.g., a drawing that includes the information from the ESN and the required approval signatures).

3.2 Equipment Requiring an Engineering Safety Note

The documentation guide for ESNs is shown in Fig. 1. The following vessels and systems require an ESN unless listed as “ESN Exempt”:

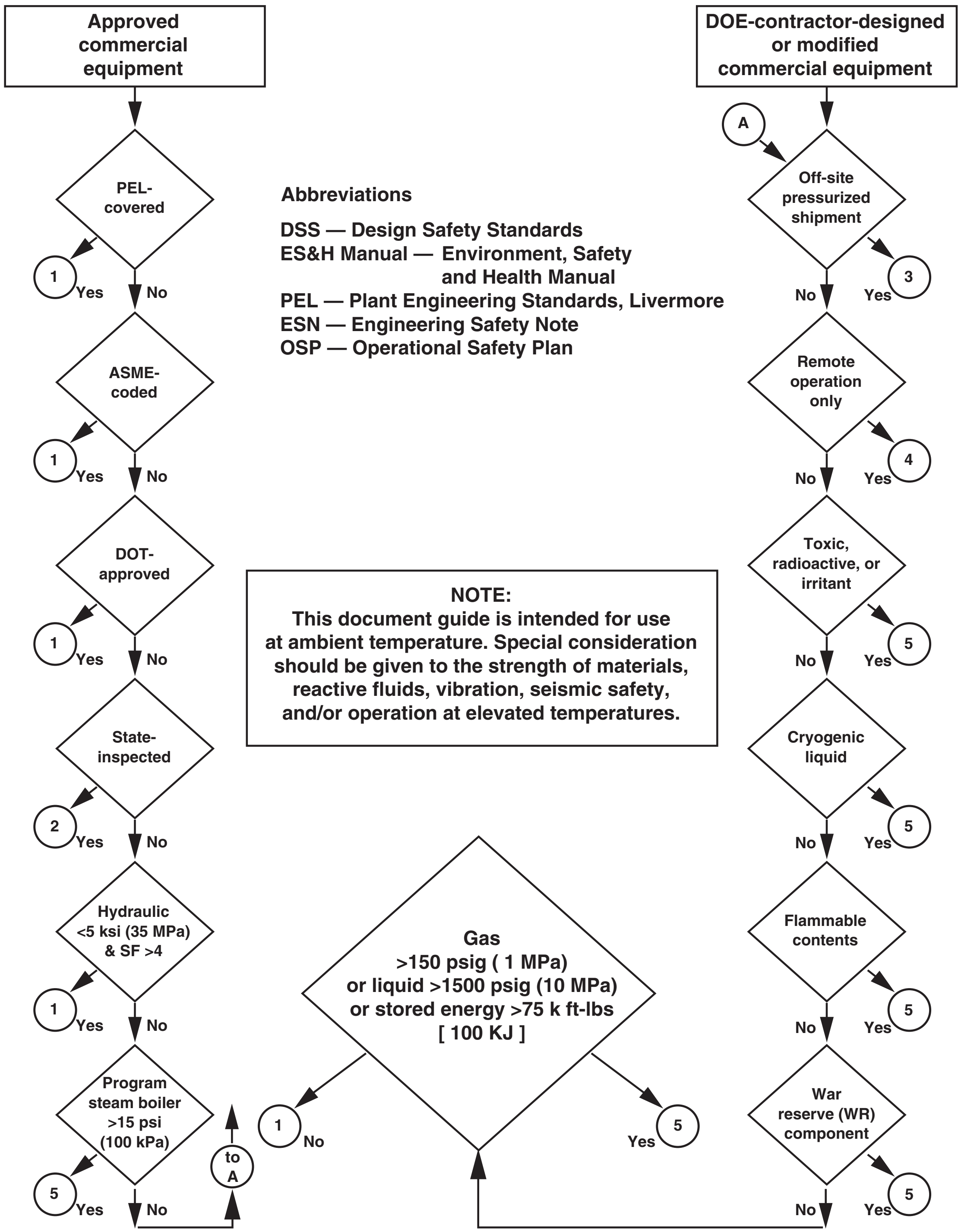
- All manned-area vessels at any gas pressure that contain over 75,000 ft-lb (100 kJ) of isentropic energy. This includes ASME-coded vessels that have been modified structurally.
- All manned-area vessels and systems at gas pressures over 150 psig (1 MPa gauge) and liquid pressures over 1500 psig (10 MPa gauge). Unmodified, commercially manufactured hydraulic systems with a safety factor of at least 4 do not require an ESN unless their working pressure exceeds 5000 psig (34.5 MPa).
- All programmatic steam boilers operating at over 15 psig (100 kPa gauge). Operation of these types of equipment must comply with 8 CCR §§ 450–560.
- All manned-area systems that confine a hazardous material at less than the above-specified limits when required by an OSP.
- All manned-area vessels or systems used with cryogenic fluids.

4.0 Design Controls for Pressure Vessels

The criteria listed in this section apply to pressure vessels used for manned-area operations. For remote operations, the extent to which these criteria apply depends on the required functional reliability.

4.1 Design Criteria

- Use a safety factor of 4 based on the known or calculated failure pressure of the vessel or ultimate strength of the material when designing for normal manned-area operation. Use a higher factor if an operation involves detrimental conditions, such as vibration, corrosion, shock, or thermal cycling.
- Never use a safety factor less than 4 when designing a vessel for manned-area operation unless the design conforms to the ASME code or to the requirements listed in Section 4.9 of this document and is approved by the division leader.
- Have the Deputy Associate Director for Mechanical Engineering approve any manned-area vessel design that is based on a safety factor of less than 3.
- Design any vessel or system containing hazardous materials such that the contained fluid leak rate will not pose a hazard to personnel.



1. No ESN or OSP is required (documented hazards are low). Go to “A.”
2. No ESN or OSP is required, but notify PE M&O and facility management. Go to “A.”
3. Approved DOT and / or DOE shipping containers that comply with LLNL shipping requirements.
4. No ESN is required for the pressure equipment. Comply with Chapter 4 of the ME Design Safety Standards. An OSP may be required (see Document 3.3, “Operational and Facility Safety Plans,” in the *ES&H Manual*.
5. An ESN, and / or OSP is required, or you should follow PEL-M-13200.

Figure 1. LLNL documentation requirements for pressure equipment.

4.2 Material Selection

- Select materials that remain **ductile** throughout the working temperature range of the vessel. If you cannot avoid using a **brittle** material for the body of a manned-area pressure vessel, your Department Head must sign the ESN.
- Select materials that are compatible with the liquid or gas to be contained in the vessel.
- Beware of **hydrogen embrittlement**. High-pressure hydrogen gas drastically degrades the ductility of highly stressed, high-strength pressure vessel materials. This problem can be solved using either one, or both, of the following methods:
    1. Use lower-strength vessel materials such as type 304, 316, 321, 347, or 21-6-9 stainless steel; 2024 or 6061 aluminum alloy; oxygen-free copper; phosphor bronze; beryllium copper; or other materials recommended by a recognized expert in the field or through a peer review.
    2. Include an inner liner (or bladder vessel) made of one of these hydrogen-resistant materials. When designing such a liner, be sure that it will withstand working and testing stresses. Consider positively venting the liner / body interspace so that any hydrogen that penetrates the liner cannot subject the high-strength vessel body to high-pressure hydrogen. Provide means for periodic verification that the vent path is open to the atmosphere.
- Consider the **creep** characteristics of the material. This is particularly important when the pressure is to be contained for extended periods at elevated temperatures.
- Make sure the vessel material is of acceptable **fracture toughness** throughout its working temperature range. Various test methods may be employed to make this toughness evaluation, e.g., Dynamic Tear Test (ASTM E604-77 or latest version), Plane Strain Fracture Toughness of Metallic Materials (ASTM E399-78 or latest version), or J-Integral Test. In addition, Charpy impact tests (ASTM E23-72, 78) should be included to check material variability. Charpy impact values of less than 22 ft-lb (30 J) or “equivalent K_{Ic}” (via J_{Ic} Dynamic Tear Energy or K_{Ic} tests) values of less than 100 ksi-in.$^{1/2}$ are often found to be unacceptable for manned-area operation; however, the actual required toughness values should be determined to ensure safe operation of the vessel. The required critical crack size should provide for “leak-before-break” with a safety factor of 4 on the flaw dimension (not on fracture toughness). For pressure vessels with wall thicknesses greater than 2 in. (50 mm) and working pressures over 14.5 psig (100 kPa), specify impact testing of vessel specimens at the lowest vessel working temperature or at 20°F (–7°C), whichever value is lower.

- Confirm the **material's identity** by verifying it to be of a particular specification using x-ray diffraction, chemical analysis, metallography, radiography, or sample testing, as required.

Materials listed in Table 1 of this document are normally satisfactory for pressure-vessel fabrication. The strength values apply between –20°F (–30°C) and 200°F (95°C). At working temperatures below ambient, there is a possibility of brittle behavior; at temperatures over 200°F (95°C), reduction in strength usually becomes significant. The tabulated information is from Refs. 1–4.

Refer to Section 5.4 (Fracture Critical Components) of the ME *Design Safety Standards*[5] for design and documentation requirements. Questions about the fracture integrity of the vessel should be directed to the Mechanics of Materials Group, Nondestructive and Materials Evaluation Section, Manufacturing and Materials Engineering Division (MMED).

4.3 Design Considerations

- Specify that all purchase-fabrication welding be done by certified ASME welders in accordance with the approved ASME Boiler and Pressure Vessel Code.[1]
- Avoid longitudinal welds in vessels less than 6 in. (0.15 m) in diameter. Seamless tubing or pipe, or bar stock, is usually available in these smaller diameters.
- Avoid stress concentrations. This is most critical when vessel material elongation or fracture toughness is relatively low.
- Adjust the design and the allowable stresses to compensate for environmental conditions such as vibration, cycling, temperature fluctuation, shock, corrosion, and extreme thermal operating conditions.
- Specify inspection by appropriate nondestructive detection methods, such as radiographic, ultrasonic, dye penetrant, and magnetic particle inspection, when designing a high-strength, high-pressure vessel. Specify appropriate ultrasonic inspection of all manned-area pressure vessels with wall thicknesses over 2 in. (50 mm). Maximum permissible defects should be based on the capability of the vessel material to resist crack growth under the specified operating conditions. Contact the subject-matter expert for assistance with properly specifying ultrasonic inspection.

Table 1. Specifications of materials and alloys for use in pressure vessel.

Grade or Type	Hardness (Rockwell)	Minimum ultimate tensile strength		Minimum yield strength		Remarks
		ksi	(MPa)	ksi	(MPa)	
Low-Carbon Steels						
ASTM SA-30	—	55	(380)	30	(210)	
ASTM SA-129	—	40	(275)	22	(150)	
ASTM SA-201	—	55	(380)	30	(210)	
ASTM SA-299	—	75	(500)	40	(275)	
ASTM SA-414	—	45	(310)	24	(165)	
Low-Alloy Low-Carbon Steels						
ASTM SA-202	—	75	(500)	45	(310)	
ASTM SA-203	—	65	(450)	37	(255)	
ASTM SA-225	—	70	(485)	40	(275)	
ASTM SA-353	—	90	(620)	60	(415)	
ASTM SA-357	—	60	(415)	30	(210)	
ASTM SA-387	—	60	(415)	35	(240)	
USS “T-I”	—	115	(800)	90	(620)	
HY 80	—	105	(~725)	90	(620)	
HY 100	—	125	(~860)	110	(760)	
HY 180	—	200	(~1380)	180	(1240)	
Alloy Steels (all tempered at 700˚ (370˚C) or higher)						
4130	25–30 R_C	125–145	(860–1000)	103	(710)	
32	32–36 R_C	150–170	(1035–1170)	132	(910)	0.3 in (7.5 mm) thick
8630	25–30 R_C	125–145	(860–1000)	103	(710)	max.
	32–36 R_C	150–170	(1035–1170)	132	(910)	0.3 in (7.5 mm) thick
4340	25–30 R_C	125–145	(860–1000)	103	(710)	max.
	32–36 R_C	150–170	(1035–1170)	132	(910)	Beware of low
	39–43 R_C	180–200	(1240–1380)	163	(1125)	fracture toughness
	43–46 R_C	200–220	(1380–1515)	175	(1210)	
Titanium Alloys (beware of brittle welds)						
TI-5 A1-2.5 Sn	—	115	(800)	90	(620)	RS-110C, A-110 AT
TI-6 A1-4 V	—	130	(900)	120	(830)	RS-128, C-120 AV
Austenitic Stainless Steels (resistant to hydrogen embrittlement)						
	annealed	95–100	(655–690)	45	(310)	
21-6-9 VIM / ESR	96 R_B	112	(770)	68	(470)	
(ASTM A-276)	34 R_C	145	(1000)	130	(900)	
304	70–90 R_C	70–90	(485–620)	25–50	(170–345)	Enhanced properties
	10–35 R_C	100–150	(700–1240)	50–150	(345–1035)	result from warn
316	70–85 R_C	75–90	(500–620)	30–60	(210–415)	high-energy rate
	10–30 R_C	100–150	(700–1035)	50–125	(345–860)	forging
321, 347	70–90 R_C	75–90	(500–655)	30–55	(210–380)	
	10–35 R_C	100–150	(700–1035)	50–125	(345–860)	

(1) See UW-12(a) and UW-51.

(2) See UW-12(b) and UW-52.

(3) The maximum allowable joint efficiencies shown in this column are the weld joint efficiencies multiplied by 0.80 (and rounded off to the nearest 0.05), to effect the basic reduction in allowable stress required by this Division for welded vessels that are not spot examined. See UW-12(c).

(4) Joints attaching hemispherical heads to shells are excluded.

- Prepare a Fracture Control Plan for all gas-pressure vessels with wall thicknesses over 2 in. (50 mm) that are to be operated in a manned area. These vessels should be periodically monitored using appropriate nondestructive inspection techniques to assure that previously undetectable, undetected, and detected cracks are not approaching critical size. Contact the subject-matter expert for assistance. A plan should be prepared for vessels with thinner wall thicknesses wherever radioactive, toxic, explosive, or flammable materials are involved.
- When specifying welding of pressure vessel components, consider the following:
  - — Checking a weld cross section for toughness, because a weld might be brittle and welding might embrittle the material in the heat-affected zone.
  - — Including realistic joint efficiencies in calculations (see Ref. 1, Table UW-12), because a weld might not penetrate to the full thickness of the parent material.
  - — Including the reduced properties of the heat-affected zone when calculating the overall strength of the vessel, because welding normally anneals the material in this zone.
  - — Consulting with a welding or materials expert when planning to weld a vessel that will contain a high-pressure hydrogen gas, because welding reduces resistance of some materials to hydrogen embrittlement.
- Use a realistic MAWP as a basis for design calculations. Select an MAWP that exceeds the highest anticipated MOP by 10–20% (see Fig. 2). This permits proper relief protection against overpressure without degrading the dependable leak-tight function of the vessel at its operating pressure.
- Provide positive protection against overpressure by installing a relief device set at a pressure not exceeding the MAWP of the vessel.
- Design all barricades for remote-operation pressure systems in accordance with the requirements in Ref. 5.

4.4 Calculation Guide for Ductile Vessels

Equations (1), (2), (3), and (4) are based on the maximum allowable circumferential (or hoop) stress, *not* on the true combined stress condition of the vessel. The actual stress near a weld joint or in any area of stress concentration will be considerably higher than the “average” stress that results from applying these equations. However, proper application of these equations will result in a vessel of ASME code-equivalent safety. (See Appendix C for help in locating information in the ASME Boiler and Pressure Vessel Code.[1] For additional design information, see Section 8.0, “References.”

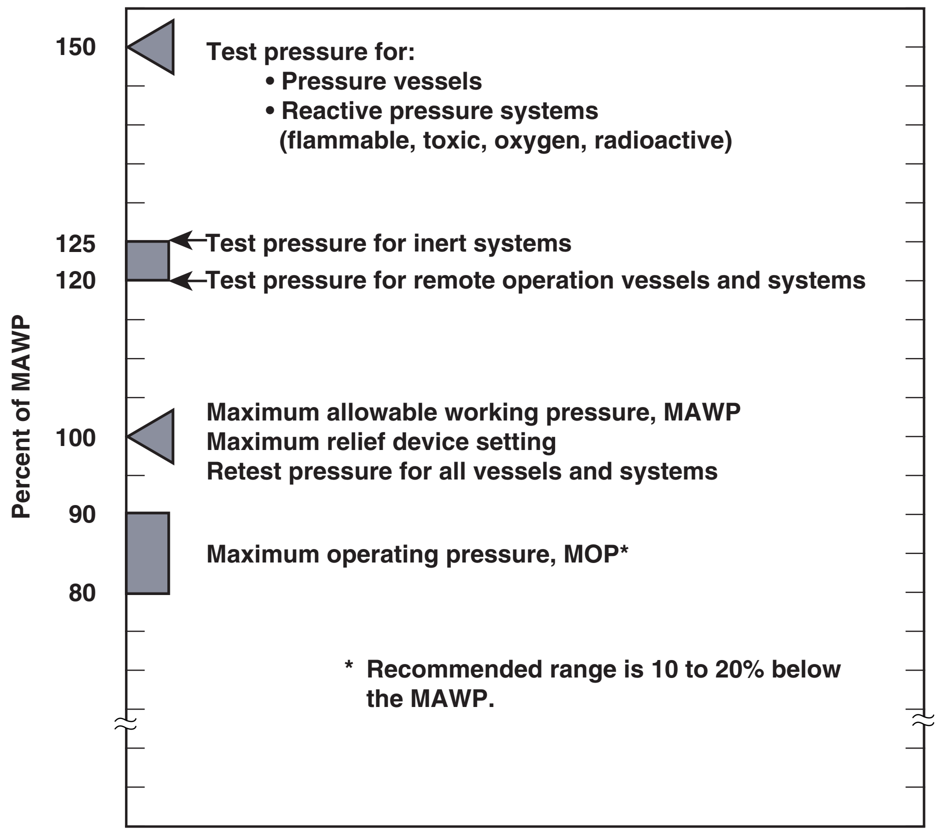


Figure 2. Relationships between test pressures, the MAWP, and MOP.

The following notations apply to the equations given in this section:

C = attachment coefficient (see Fig. 3).

d = internal diameter of vessel, in inches or meters.

E = joint efficiency factor, usually 1, except for welded vessels. (See Ref. 1, Table UW-12 in Appendix F, for welded joint efficiencies.)

h_G = radial difference between the bolt circle and the pressure-seal circle on a bolted-end enclosure. inches or meters.

k = ratio of specific heats, c_p / c_v.

P = maximum allowable working pressure, psig or Pa.

r_i = inner radius, inches or meters.

r_o = outer radius, inches or meters.

R = ro / ri.

Sa = allowable stress of material, psi or Pa.

SF_u = safety factor based on ultimate strength of the material.

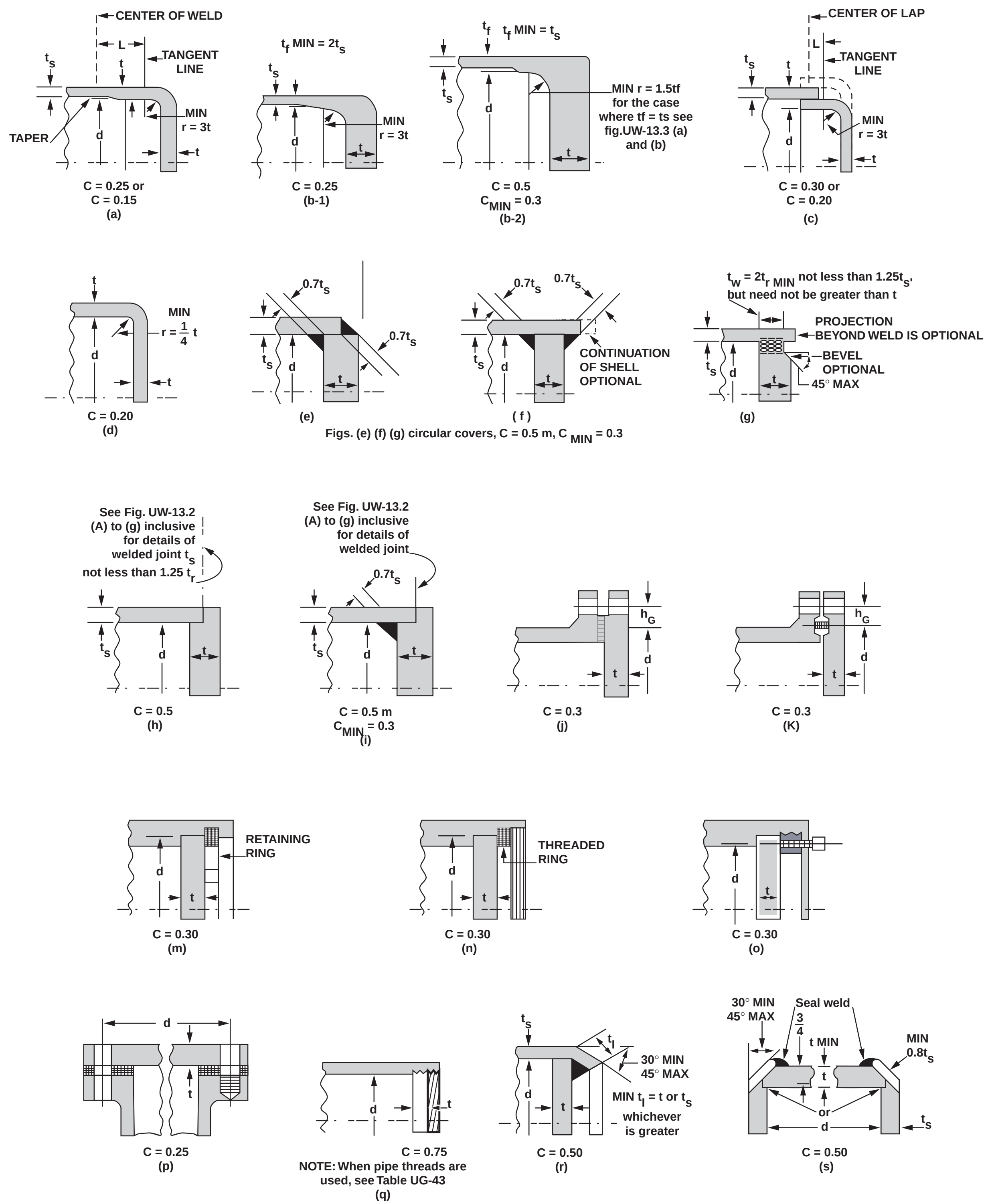


Figure 3. **Some acceptable types of unstayed flat heads and covers. The symbol "m" is the ratio tr/ts, where tr is the required shell thickness, exclusive of corrosion allowance. Designs, other than those shown, that meet the requirements of UG-34 are also acceptable. (This figure was reproduced from Fig. UG-34 in Ref. 1, with the permission of the ASME.)**

σ_u = ultimate strength of material, psi or Pa.
σ_y = yield strength of material, psi or Pa.
t = wall thickness, inches or meters.
U = energy, ft-lb or joules.
v = volume, in.^{3} or m^3.
W = total bolt load for circular heads, lb or N. (Pressure force plus required gasket sealing force.)

For **thin-wall vessels**, where R is less than 1.1, use Eq. (1) or (2) to calculate p (the MAWP) (Ref. 6, chapter 12).

For cylinders,

$$P = \frac{S_a Et}{r_i} \text{ or } \frac{\sigma_u Et}{SF_u r_i}. \tag{1}$$

For spheres,

$$P = \frac{2S_a Et}{r_i} \text{ or } \frac{2\sigma_u Et}{SF_u r_i}. \tag{2}$$

For **medium-wall vessels,** where R is between 1.1 and 1.5, use Eq. (3) or (4) to calculate the MAWP (Ref. 1, par. UG-27).

For cylinders,

$$P = \frac{S_a Et}{r_i + 0.6t} \text{ or } \frac{\sigma_u Et}{SF_u(r_i + 0.6t)}. \tag{3}$$

For spheres,

$$P = \frac{2S_a Et}{r_i + 0.2t} \text{ or } \frac{2\sigma_u Et}{SF_u(r_i + 0.2t)}. \tag{4}$$

For **thick-wall vessels,** where R is between 1.5 and 2.0, use Eq. (5), (6), (7), or (8) to calculate the MAWP.

For cylinders,

$$P = S_a \frac{\left(r_o^2 - r_i^2\right)}{\left(r_o^2 + r_i^2\right)} \text{or} \frac{\sigma_u\left(r_o^2 - r_i^2\right)}{SF_u\left(r_o^2 + r_i^2\right)}. \tag{5}$$

$$P = \frac{2\sigma_y}{\sqrt{3}SF_u}\left(2 - \frac{\sigma_y}{\sigma_u}\right) \ln R. \tag{6}$$

For spheres,

$$P = 2S_a \frac{\left(r_o^3 - r_i^3\right)}{\left(r_o^3 + 2r_i^3\right)} \text{or} \frac{2\sigma_u\left(r_o^3 - r_i^3\right)}{SF_u\left(r_o^3 + 2r_i^3\right)}. \tag{7}$$

$$P = \frac{2\sigma_y}{SF_u}\left(2 - \frac{\sigma_y}{\sigma_u}\right) \ln R. \tag{8}$$

For **thick-wall vessels,** where R is over 2.0, use Eqs. (5) and (7) only to calculate the MAWP.

Medium- and thick-wall vessels of certain materials may also be designed in accordance with the rules in Section VIII, Division 2, of the ASME Boiler and Pressure Vessel Code[1] and the requirements in Section 4.9 (SF-3 Pressure Vessels) of this document.

For **flat, circular end-closures,** use Eq. (9) or (10) to calculate the required thickness. (See Ref. 1, par. UG-34, and Fig. 3). If no bending moment is imposed on the end-closure when securing it (i.e., welded, intergral, ring-retained; see Fig. 3[a through i] and 3[m through s]), then use

$$t = d\sqrt{\frac{CP}{S_aE}} \text{ or } d\sqrt{\frac{CSF_uP}{\sigma_uE}}. \tag{9}$$

If a bending moment is imposed on the end-closure when securing it (i.e., bolted; see Fig. 3[j] and [k]), then use

$$t = d\sqrt{\frac{CP}{S_aE} + \frac{1.9\ Wh_G}{S_ad^3E}} \text{ or}$$

$$t = d\sqrt{\frac{CSF_uP}{\sigma_uE} + \frac{1.9\ Wh_G\ SF_u}{\sigma_uEd^3}}. \tag{10}$$

Note: Refer to par. UG-34 of Ref. 6 for special calculation requirements.

The analysis described above only addresses hoop stress in a vessel wall and the thickness of end closures. Any number of other design features could be critical to safe design of a pressure vessel. These include the shear stress in threads, the tensile strength of bolt cross-sections, the strength of weldments, and the effect of vessel openings, nozzles, and supports. Therefore, a thorough analysis should be performed for these features if they are included in the vessel design.

For other vessels, such as multi-wall cylinders and other end-closure designs, see the references at the end of this document. Where stresses in a large high-pressure vessel appear to be complex or excessive, contact a qualified applied mechanics authority for assistance with performing a finite element analysis.

4.5 Stored Energy

Calculate the *energy* contained in the fully pressurized vessel and include the calculation in the ESN. Compare this value with the 3.42×10^6 ft-lb (4.63×10^6 J) potential energy of 2.2 lb (1 kg) of TNT.

For example, using Eq. (11), a fully charged, standard size 1 cylinder of nitrogen gas contains energy equivalent to about 0.5 lb (0.25 kg) of TNT. This calculation is based on reversible adiabatic (isentropic) expansion of the confined gas. Note that if pressure (p_1 and p_2) and volume (v_1) are expressed in megapascals and cubic centimeters, respectively, then the energy (U) is in joules (see Ref. 7, p. 4-25 for more details).

$$U = \frac{P_1 v_1}{k-1}\left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}\right]. \tag{11}$$

Note: k = 1.66 for He gas;
k = 1.41 for H_2, O_2, N_2, and air (from Ref. 7, p. 4-25).
P_1 = Vessel pressure
P_2 = Atmospheric pressure

For the same volume charged with water to the same pressure, the stored energy is considerably less. For this case, Eq. (12) may be used to determine the liquid stored energy content.

$$U = \frac{1}{2}\left(\frac{P_1^2 v}{B}\right), \tag{12}$$

where B = Liquid bulk modulus, psig,
= 300,000 psig for water.

This calculation yields a value of 1,742 ft. lb. (0.51 gms of TNT).

4.6 Testing

- All LLNL-designed or operated pressure vessels that require an ESN must be remotely pressure tested. Whenever practical, take pressure vessels to the ME High-Pressure Test Facility for pressure testing.

- Hydrostatic test (preferred) or gas test all manned-area pressure vessels at 150% of their MAWP. If the vessel body material has a yield strength less than about 55% of its ultimate strength (as with annealed 300 series stainless steel), use the equation on p. 69 of Ref. 8 (the Maximum Energy of Distortion Theory) to make sure that the combined stresses at 150% of the MAWP do not exceed the yield strength of the body material. If they do, reduce the test pressure accordingly (but do not reduce below 125% of the MAWP) and include the supporting calculation in your ESN. (See Appendix B for a sample calculation and Fig. 2 for the relationships between different pressures.)
- Hydrostatic test or gas test all remote-operation pressure vessels at 125% of its MAWP unless your division leader specifically approves the use of a different test pressure.
- If extreme conditions are involved in vessel operation, simulate these conditions during testing, or if simulation is impractical, consider the weakening effect of these conditions when assigning the test pressure. For instance, if it is not practical to test a high-temperature, high-pressure vessel at its working temperature, then test it at 150% of its MAWP times the ratio of its allowable stress at the test temperature to its allowable stress at the maximum working temperature.

4.7 Retesting and Inspection

The pressure inspector performs pressure inspections and records any findings on Form LL-3586. The Responsible Individual then signs the completed form and sends it to the LLNL pressure safety manager for permanent recordkeeping. The vessel or system is then tagged with the appropriate pressure label (Figs. 4, 5, or 6).

All pressure vessels and systems designed for operation at LLNL that require documentation shall be pressure tested remotely before being operated in a manned area. Once tested, an LLNL pressure-tested label shall be attached to the pressure vessel or system. Documented and labeled pressure vessels or systems and their integral pressure-relief devices shall be maintained by the Responsible Individual and inspected by a qualified independent LLNL pressure inspector every three years as recommended by NBIC. Inspection intervals for pressure vessels will be determined using in-service inspection criteria in the NBIC inspection code. Depending on the type of vessel service, the intervals may range from two years to a maximum of 10 years. Relief devices on pressure vessels shall be inspected every 3 years. In addition, pressure systems and vessels will be reinspected whenever they are disassembled and moved or redesigned, or when the application changes, even if the working pressure is reduced.

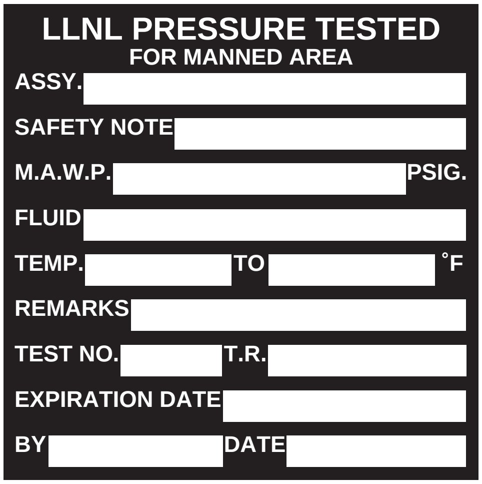


Figure 4. LLNL pressure tested label for manned-area operation (silver on black).

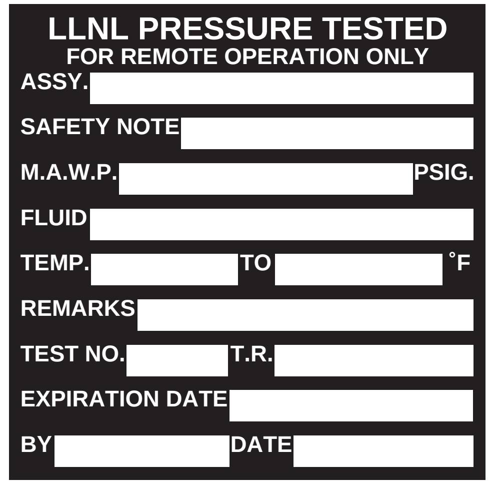


Figure 5. LLNL pressure tested label for remote operation only (silver on red).

Figure 6. Remote operation label (silver on red).

4.8 Expansion and Compression of Gases

The ideal gas law and empirical data relating to the expansion and compression of gases are generally in fair agreement in the low- and intermediate-pressure ranges. This agreement varies according to the gas in question. The Amagat chart (Fig. 7) is provided as a means for calculating the expansion or compression of nitrogen, helium, and hydrogen at a constant temperature of 25°C. Similar information for ammonia, carbon dioxide, carbon monoxide, nitrogen, air, argon, and several other gases can be found in Ref. 9.

4.9 SF-3 Pressure Vessels

The design of pressure vessels for manned-area operation normally requires a safety factor of at least 4 based on the known or calculated failure pressure of the vessel or the ultimate strength of the material. For certain special applications, designs using a safety factor as low as 3 are warranted and can be approved by the division leader. The division leader shall appoint qualified personnel to perform a peer review before approving the vessel. Pressure vessel designs involving brittle materials or with a safety factor less than 3 require a peer review and approval by the ME Deputy Associate Director.

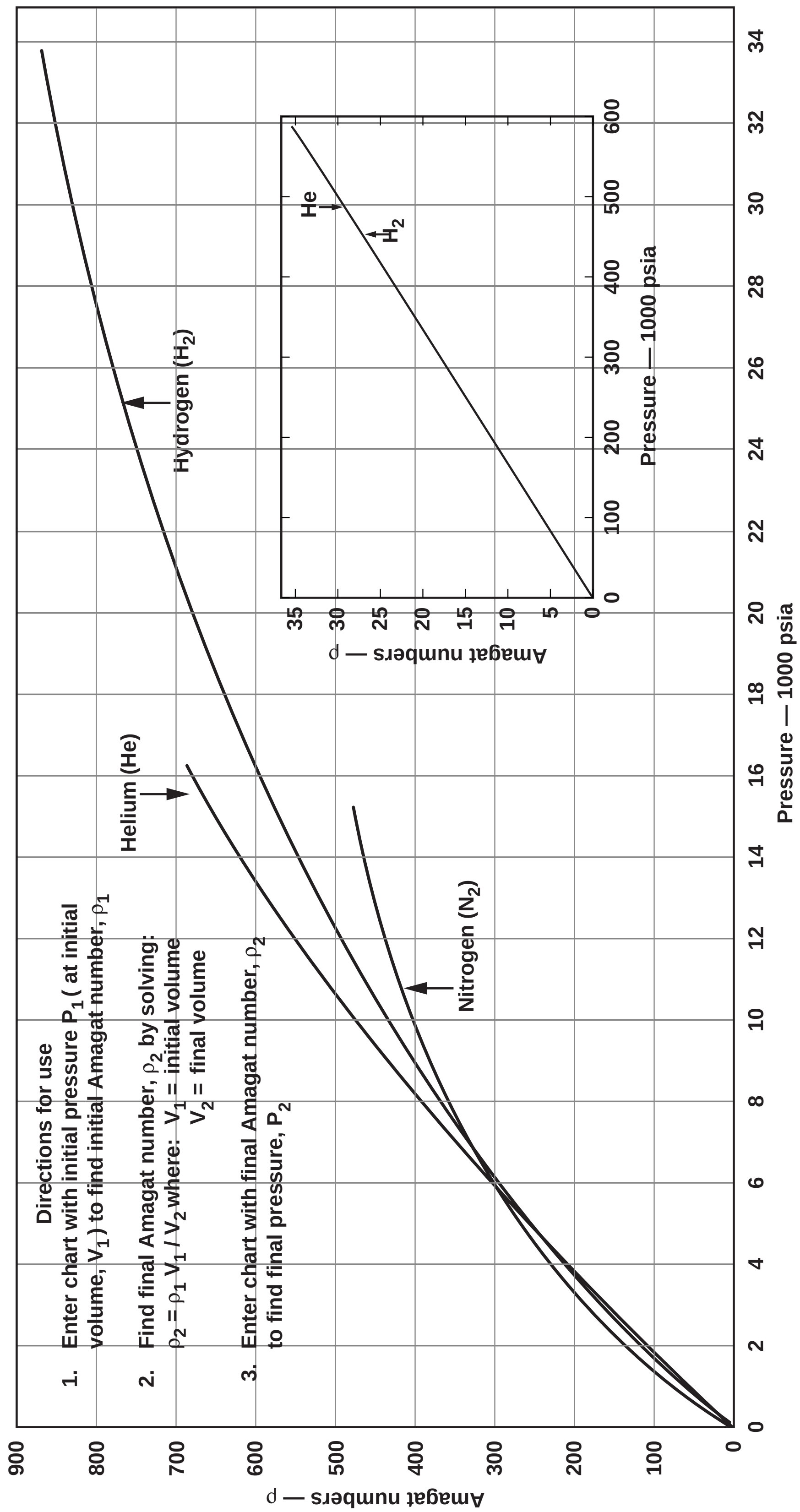


Figure 7. Amagat chart.

4.9.1 Materials

Select a ductile material that will have at least the following properties in the final heat-treated or work-hardened condition, and include a copy of the certified chemical analysis in the ESN:

1. Ultimate tensile and yield strengths equal to or exceeding those used in the vessel calculations.
2. Reduction of area of at least 40%.
3. Percent elongation of at least 15%.
4. A complete fracture evaluation and proper design selection to ensure “leak-before-break” criteria are met.
5. Demonstrated compatibility with the pressure media, or protection from the pressure media by such means as a compatible liner / end closure or bladder vessel.

4.9.2 Tensile Testing

Tensile specimens of the heat-treated or work-hardened material from each lot (material from the same heat that is processed identically at the same time and under the same conditions) shall be tested to confirm compliance with the first three material requirements listed in Section 4.9.1. At least three reliable test results shall be numerically averaged to determine compliance with each requirement. Specimens shall be taken from locations and orientations of maximum calculated stress. Specimens for large, thick-wall cylindrical and spherical designs shall be transversed and, where possible, should be taken from outer-, inner-, and mid-wall locations. Refer to Section 4.9.5 (Welded Vessels) of this document for tensile testing requirements for structural welds.

4.9.3 Toughness Testing

At least one specimen (but preferably three or more) from each lot of material shall be tested to confirm compliance with the fourth requirement in Section 4.9.1 (Materials) of this document. To meet this requirement, the material shall comply with the criteria in Section 4.2 (Material Selection) of this document, except that a safety factor of 3 will be accepted on a “through-the-thickness” flaw size. Any deviation from this requirement shall have a signed statement from the division leader and approval by the Mechanics of Materials Group, Nondestructive and Materials Evaluation Section, Manufacturing and Materials Engineering Division (MMED) personnel.

4.9.4 Compatibility

A statement affirming compliance with the fifth requirement in Section 4.9.1 shall be included in the ESN, including copies of any supporting certification.

4.9.5 Welded Vessels

The following additional requirements apply to all certified vessels containing structural welds:

- All welding shall be done by qualified LLNL workers, DOE contractors or subcontractors, or DOE production facility welders in accordance with approved welding procedures, or by certified ASME welders in accordance with Section IX of the ASME Boiler and Pressure Vessel Code.[1]
- Only welds done by the TIG, MIG, EB, EBCWF (electron beam, cold wire feed), shielded metal arc, and submerged arc methods are permitted.

The procedure for confirming the assumed efficiency of welds for each lot (including typical degradation of physical properties in the heat-affected zone) requires that a facsimile of each welded section be welded at the same time and under the same conditions as the parent weld. Each welded section shall be metallographically sectioned, and three tensile specimens shall be prepared and tensile tested. At least one prototype vessel shall be burst tested. Test results shall verify the weld efficiency used in the final vessel design calculation.

4.9.6 Nondestructive Testing

Each welded vessel shall be 100% radiographed or ultrasonically inspected, and all structural welds shall be 100% dye-penetrant or magnetic-particle inspected, as required, to confirm weld quality, depth of weld penetration, and absence of unacceptable voids, cracks, and inclusions. Where practical, a radiograph window (a small but detectable annular groove that will be fused by welding of acceptable penetration) should be designed into girth-weld joints to facilitate the determination of weld penetration.

4.9.7 Pressure Testing and Labeling

Each finished vessel shall be pressure tested at 150% of the MAWP unless the Maximum Energy of Distortion Theory analysis of combined stresses indicates that the vessel will yield at this test pressure. In this case, testing shall be at a pressure slightly below theoretical yielding but not less than 125% of the MAWP.

The LLNL pressure-tested label (see Fig. 4 and 5) shall be marked “SF-3” in the remarks section. Refer to Document 18.3, “Pressure Testing,” in the *ES&H Manual* for specific testing requirements.

4.10 War Reserve Vessels

This section covers documentation and handling requirements for LLNL war reserve pressure vessels and assemblies. These pressure vessels are normally designed and fabricated at other DOE / DoD facilities and are usually pressurized before arrival at LLNL.

4.10.1 Documentation

If the subject vessel or assembly would require a Mechanical Engineering ESN if it were designed locally, an ESN is required. This ESN must be prepared, reviewed, and approved by the same technical and management levels required of other “ESN-required” vessel designs.

4.10.2 Design Criteria

War reserve vessels fabricated by high-energy-rate forging (HERF) from 21Cr–6Ni–9Mn, 304L, or JBK75 stainless steel of work-hardened yield strength less than 122,000 psig are considered safe for manned-area operation, provided the burst safety factor is at least 3. War reserve vessels with a lower safety factor require approval by an LLNL Department Head or Associate Director.

4.10.3 Handling

Before handling a war reserve vessel in a manned area, the responsible designer or Responsible Individual must verify that the equipment is not pressurized at over its room temperature MAWP. Certification of the charged pressure from the supplier is normally required. Identification of the person responsible for the charge pressure and the method for confirming it must be included in the ESN. If the vessel or assembly cannot be approved for manned-area operation, it must be enclosed in a containment vessel (see Section 5.0 of this document for details) or have the charge pressure reduced to an acceptable level before transport to LLNL.

5.0 Design Controls Gas-Pressure Containment Vessels

The requirements in this section apply to equipment used as protective enclosures for gas-pressurized vessels, including those that contain toxic, radioactive, corrosive, or flammable materials. These types of equipment must be designed to protect personnel from the hazards of pressure-vessel failure (e.g., blast pressure and flying fragments). If hazardous materials could escape from the contained vessel (in case of media leakage), the containment vessel must be designed to prevent subsequent leakage to the atmosphere.

5.1 Special Shipping Requirements

Only DOT- or DOE-approved containers shall be used for offsite shipment of pressure vessels containing radioactive materials. See Ref. 10 for DOT shipping regulations; Ref. 11 (or Materials Management) for DOE requirements; and Ref. 12 for information about gas-sampling cylinders where only small quantities of radioactive materials are involved in a shipment.

5.2 Design Safety Factors

If the contained pressure vessel is of ductile material and has been approved by LLNL for a manned-area MAWP of at least the maximum pressure to which it could be subjected inside the containment vessel, the containment vessel shall be designed to an ultimate or burst safety factor of at least 4. If the **contained** pressure vessel has *not* been LLNL-approved for a manned-area MAWP of at least the maximum pressure to which it is to be subjected inside the containment vessel, the containment vessel for manned-area operation shall be designed to an ultimate or burst safety factor of at least 8.

5.3 General Design Requirements

The following requirements apply to all gas-pressure containment vessels, including those designed, specified, or used by LLNL personnel, that will contain toxic, radioactive, corrosive, or flammable materials.

- Design the containment vessel using the appropriate safety factor specified in Section 5.2 (Design Safety Factors). Base the design on the maximum equilibration pressure expected if the **contained** pressure vessel is heated to the highest temperature expected within the containment vessel or to 130°F (55°C), whichever value is higher.
- In selecting materials of satisfactory fracture toughness, assume a minimum operating temperature of nil ductility temperature (NDT) (40°F), unless a lower temperature is required and specified.
- If offsite transportation is to be permitted, design the containment vessel to withstand the normal conditions of transport, as listed in Annex 1 of Ref. 11. This includes heat, cold, pressure, vibration, water spray, free drop, corner

drop, penetration, and compression. Annex 1 requirements also state that the contained vessel shall be mounted securely inside the containment vessel.

- Include a compound pressure / vacuum gauge for periodically monitoring the internal pressure of the containment vessel. This gauge shall be graduated to at least 120% but not over 200% of the containment vessel MAWP. The highest credible equilibration pressure is the MOP of the containment vessel.
- Include two separate valve entries for safely introducing, exhausting, monitoring, and flushing gas through separate lines.
- Include suitable covers and shields to protect all valves and gauges from damage. Cap or plug all terminal valve ports. Provide accommodations for locking or wiring valve handles closed or having valve handles removed during shipment to prevent unauthorized operation or tampering.
- If the contained vessel has not been LLNL-approved for a manned-area MAWP of at least the maximum pressure to which it could be subjected inside the containment vessel, refer to Section 5.1 of Ref. 1. Show that credible flying objects would not penetrate the containment vessel if it failed catastrophically.

5.4 Testing and Labeling

- Pressure test the containment vessel at 150% of its maximum possible equilibration pressure. To determine the maximum equilibration pressure, assume that the most energetic contained vessel specified equilibrates into the containment vessel, which is then heated to 130°F (55°C), unless a higher temperature is specified. No detectable plastic strain is permitted, as determined by measurements to within 0.001 in. (0.025 mm), both before and after testing.
- After successful pressure testing, leak check the containment vessel at the maximum possible equilibration pressure with a leak detector capable of detecting leakage of 1×10^{-8} atm cm^3 / sec. No detectable leakage is permitted.
- Specify contained vessel rupture testing of the containment vessel if necessary.
- After successful testing and leak checking, make sure the pressure inspector affixes a label to the containment vessel indicating the following:
  - — The working pressure used as the basis for the design calculation and test.
  - — A working temperature range of –20°F to 130°F (–29°C to 55°C), unless a wider temperature range is required or specified.

6.0 Design Controls for Pressure Systems

6.1 Precautions

The following precautions shall be observed when designing, installing, or operating a pressure system.

- Be sure that the MAWP and MOP are on all pressure system assembly drawings.
- Limit pressure sources to the MAWP of the lowest rated system component. Do *not* consider a pressure regulator by itself as satisfactory overpressure protection.
- When pressure sources cannot be limited to less than the MAWP of every system component, include pressure-relief devices (relief valves or rupture-disc assemblies) to protect those components that are rated at less than the system supply pressure. All gas pressure vessels used for manned-area operations must have a relief device that is set at a pressure not exceeding the MAWP of the vessel.
- Do not use the following:
  - — Steel threaded fittings at pressures over 1 MPa (150 psig) or brass threaded fittings at pressures over 0.83 MPa (125 psig) unless the stamped rating, manufacturer’s catalog, or other reference states they have a higher pressure rating.
  - — Tubing or pipe at pressures above those listed in this document unless such use is specifically covered by an approved ESN.
  - — Threaded pipe other than seamless Schedule 80 (at least) for 1.7 MPa (250 psig) steam service or 0.7 MPa (100 psig) service with water over 105˚C (220˚F).

6.2 Pipe and Tubing

Use pipe and tubing rated at or above the required MAWP. If you plan to use pipe or tubing at pressures above the listed values, include calculations in an ESN to justify your selections.

When selecting pipe or tubing, consider the following:

- Operating pressure and temperature.
- Fluid compatibility.

- Installation / maintenance requirements.
- Proper hardness.

Use the American National Standard Institute Code for pressure piping, ANSI B31.1, or a reliable reference to determine the MAWP for low- and intermediate-pressure pipe and tubing. Refer to the tables in Appendix D for pressure ratings for various pipes and tubings.

6.3 Pipe and Tube Support

- Secure all components of pressure systems.
- Support and secure hose and tubing at least every 7 feet (2 m) in manned areas. Support and secure pipes in manned areas as specified in Table 2. Locate supports to limit strain on fittings and minimize overhang at bends. Consider that pipe and tubing expand and elongate when heated and contract when cooled. Use additional supports for heavy system components.
- Use adequate machine screws (or bolts) and nuts to secure all components. Wood screws are not considered adequate.

Table 2. Suggested pipe support spacing (Ref. 13).

Nominal pipe size	Suggested Maximum Span			
	Water Service		Steam, gas, or air service	
(in)	(ft)	(m)	(ft)	(m)
1	7	2.1	9	2.7
2	10	3.0	13	4.0
3	12	3.7	15	4.6
4	14	4.3	17	5.2
6	17	5.2	21	6.4
8	19	5.8	24	7.3
12	23	7.0	30	9.1
16	27	8.2	35	10.7
20	30	9.1	39	11.9
24	32	9.8	42	12.8

NOTES:

(1) Suggested maximum spacing between pipe supports for horizontal straight runs of standard and heavier pipe at maximum operating temperature of 750°F (400°C).

(2) Does not apply where span calculations are made or where there are concentrated loads between supports, such as flanges, valves, specialties, etc.

(3) The spacing is based on a fixed beam support with a bending stress not exceeding 2,300 psig (15.86 MPa) and insulated pipe filled with water or the equivalent weight of steel pipe for steam, gas, or air service, and the pitch of the line is such that a sag of 0.1 in. (2.5 mm) between supports is permissible.

6.4 Fittings

When selecting a fitting, consider the following:

- Rated working pressure of the fitting and system.
- Compatibility and operating temperature of the fitting material with the system fluid.
- Availability of replacement units or component parts.
- Proven quality, dependability, and cost of the fitting in relation to its required performance.

Assume that all steel pipe fittings (unless otherwise marked or identified) are rated at 150 psig and all brass pipe fittings are rated at 125 psig. A fitting or valve marked “125 WOG” is good for up to 125 psig of water, oil, or gas at room temperature. A fitting marked “150” may be good for up to 275 psig of gas pressure, but it is not to be used at pressures over 150 psig unless the higher pressure rating can be proved. Refer to the manufacturer’s catalog or other in-house reference for more details.

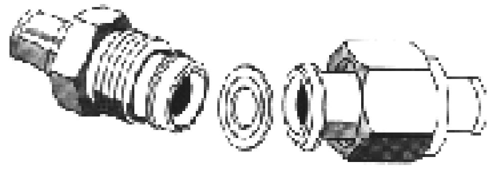
In the following text regarding fittings, the MAWP will *usually* be determined by tube size (i.e., outside diameter, o.d; inside diameter, i.d). However, if the fitting incorporates a “weaker element,” such as in a tube-to-pipe adapter, the pipe thread will usually have a lower MAWP than the tubing used. Therefore, the lower MAWP must be used.

6.4.1 National Pipe Taper Thread (NPT) Fittings

These fittings seal by interference fit and require use of sealant or lubricant. Do not interchange these fittings with National Pipe Straight thread (NPS). Forged fittings are available for MAWPs of 1,000; 2,000; 3,000; 4,000; and 6,000 psig. Never use fittings at pressures over 10,000 psig.

6.4.2 Straight-Thread (Face Seal) Fittings

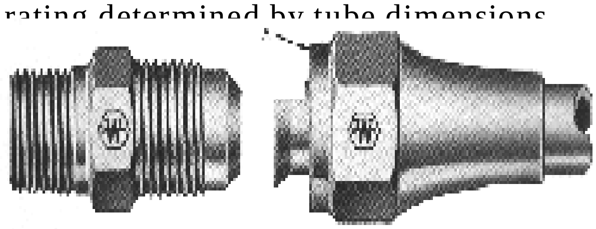
These fittings (shown below) are made of stainless steel and require a gasket or elastomer sealing device. They are used for ultra-clean vacuum and pressure systems. Refer to the manufacturer’s catalog for the working pressures.



6.4.3 Flare Fittings

There are two common types of flare fittings (shown below). The characteristics of each are listed below.

- 45° flare
  - — Two-piece. Used with copper, brass, aluminum and welded steel hydraulic tubing.
  - — Tube end flared to seal on mating part.
  - — Pressure rating determined by tube dimensions



- 37° flare
  - — Three-piece. Used with brass, aluminum, steel, and stainless steel.
  - — Pressure rating determined by tube dimensions.
  - — The minimum and maximum wall thickness for an efficient 37° flare joint are as follows:

Material: Steel, stainless steel, brass, aluminum	
Tubing o.d. (inches)	**Wall thickness (inches) (Min./Max.)**
1/8, 3/16	.010/.035
1/4, 5/16, 3/8	.020/.065
1/2	.028/.083
5/8	.035/.095
3/4, 7/8	.035/.109
1	.035/.120

Do the following to assemble flare fittings:

- Cut the tubing off squarely.
- Remove burrs and clean the tubing.
- Install gland nut and collar.
- Flare the fitting to the correct angle; use the proper tools.
- Assemble completely and tighten the fittings.
- Disassemble and check the fitting, then reassemble and retighten about 1/8 turn past finger tight. If required, refer to the manufacturer’s assembly torque specifications.

6.4.4 Flareless or Bite-Type Fittings

These fittings (shown below) are made of steel, stainless steel, or copper. The pressure seal for these fittings is achieved by a single or two-piece ferrule system that either bites or deforms the tube o.d. as the fitting is tightened.

When using flareless or bite-type fittings,

- Consider proper hardness when selecting the tubing.
- Do not interchange different manufacturer’s components.
- Determine the pressure rating by tube dimensions.

Following are the minimum and maximum wall thicknesses for an efficient bite-type joint:

Material: Steel, stainless steel, copper	
Tubing o.d. (inches)	**Wall thickness (inches) (Min./Max.)**
1/8	.028/.035
3/16	.028/.049
1/4	.028/.065
3/8	.035/.065
1/2	.049/.083
3/4	.065/.109
1	.083/.120

Do the following to assemble bite-type flareless fittings:

- Cut the tubing off squarely.
- Remove the burrs and clean tubing.
- Install the gland nut and the sleeve (or ferrules).
- Place the end of the tubing into the fitting body and tighten the gland nut until the tubing will not rotate by hand. A drop of oil on the male threads will help.
- Tighten 1-1/4 turns.
- Disassemble and check the fitting, then reassemble and retighten about 1/8 turn past finger tight.

Note: Assembly and reassembly procedures may vary between manufacturers with regard to fitting design, tube diameter, tube wall thickness, etc.

Flareless fittings made of stainless steel may be used for pressures up to 15,000 psig MAWP. Fittings for 1/16 in. and 1/8 in. o.d. tubing are standard (see Fig. 8). Flareless fittings employ a single sleeve that “clamps” onto tubing, and the gland nut will “bottom out” when the assembly is made properly.

Following are the minimum and maximum wall thicknesses for an efficient bite-type joint on higher pressure fittings:

Material: Stainless steel	
Tubing o.d. (inches)	**Wall thickness (inches) (Min./Max.)**
1/16	.017/.028
1/8	.032/.053

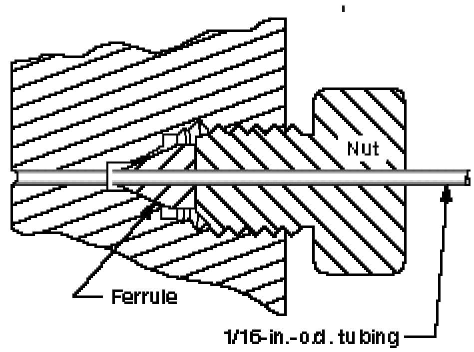


Figure 8. Typical 1/16-in. high-pressure compression fitting.

6.4.5 Coned and Threaded Connections

Coned and threaded fittings (Figs. 9 and 10) may be used to 150,000 psig MAWP depending on the manufacturer's design. Coning provides line-contact sealing, resulting in a minimal seal area. Threading locks the tube to the fitting using a collar. Fittings for 1/4-, 3/8-, and 9/16-in.-o.d. tubing are standard. The tubing and collar are left-hand threaded, and two to three threads are exposed at the tube end when the collar is screwed tightly onto a properly threaded tube. Because there are several types of coned and threaded connections, it is important that the correct tubing, collars, and gland nuts are used and are not interchanged. Special hand tools are available for coning and threading high-pressure tubing.

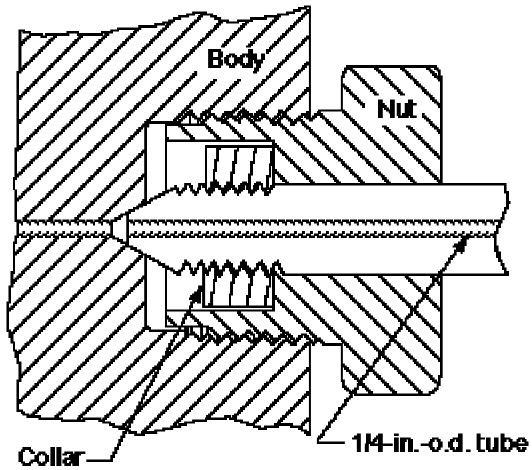


Figure 9. Typical 1/4-in. high-pressure coned and threaded fitting (60,000 psig).

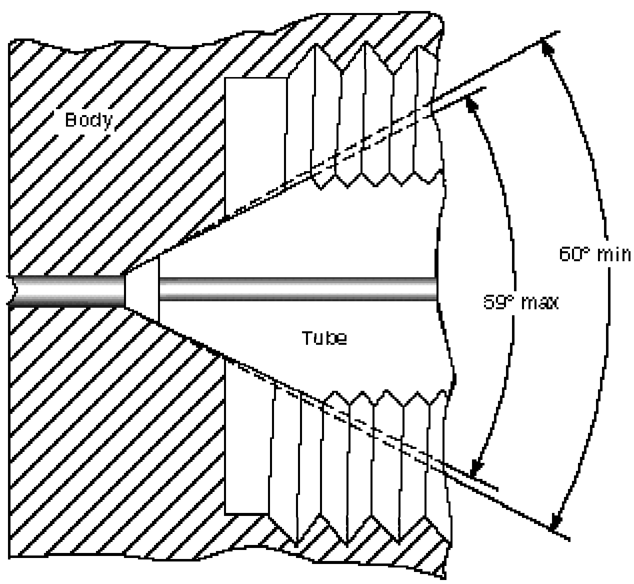
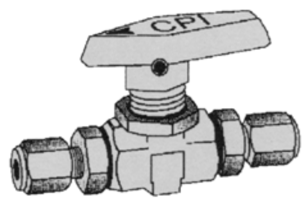
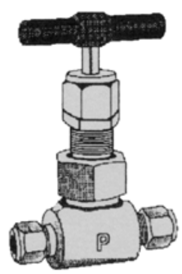


Figure 10. Typical high-pressure coned and threaded connection (to 60,000 psig).

6.5 Valves

Valves (shown below) are used to control the flow of fluids. Many types of valves are available and their applications frequently overlap.





The most common types of valves in the low- to intermediate-pressure range include ball, plug, metering, and diaphragm valves. These are available for a wide variety of applications and have various end connections. Always refer to manufacturer's catalog for specific use.

Valves in the higher pressure range (up to 150 ksi) typically employ coned and threaded connections (see Fig. 11). Nonrotating stems are commonly used to minimize leaks, and this results in a longer service life of the equipment. A variety of stem tips and body patterns are available depending on flow requirements (see Fig. 12).

Consider the following when selecting a valve:

- Operating pressure/temperature.
- Flow requirements.
- Fluid compatibility.
- Connection type and size.
- Flow pattern.
- Flow control (i.e., shut off, regulating, metering).

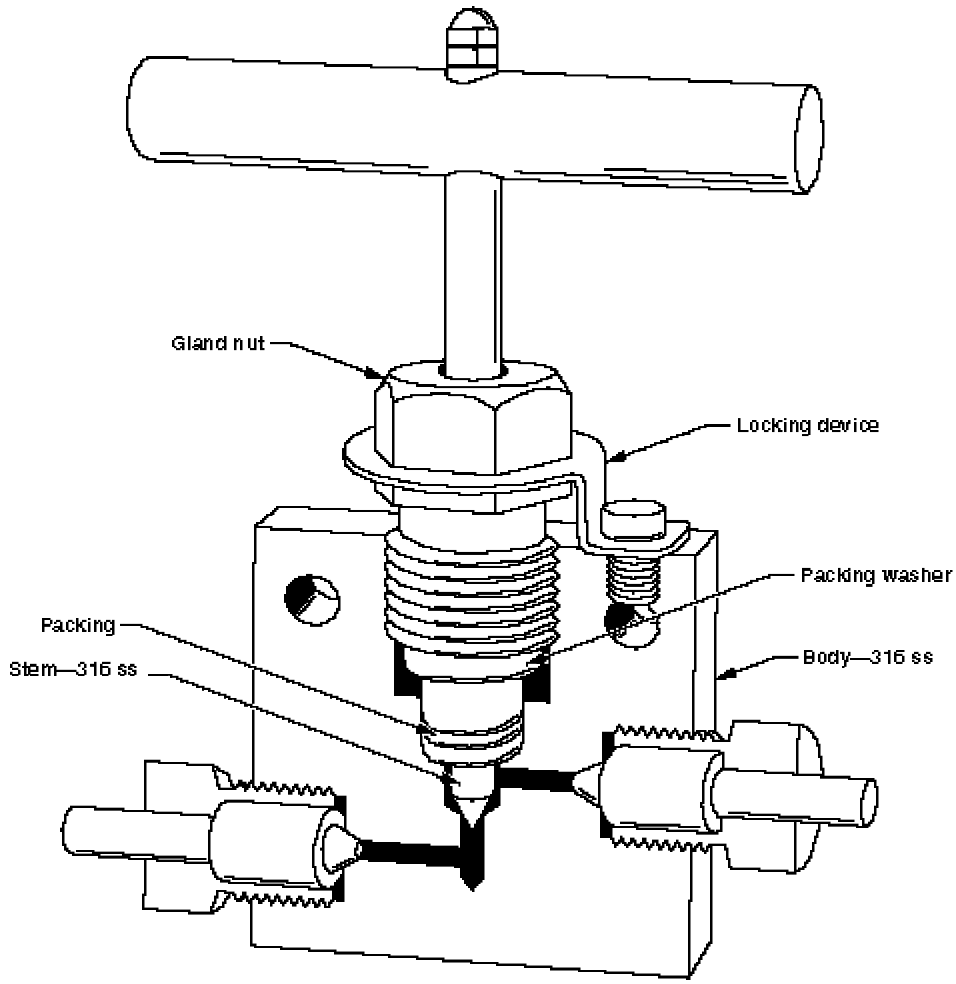


Figure 11. Type 1 high-pressure valve.

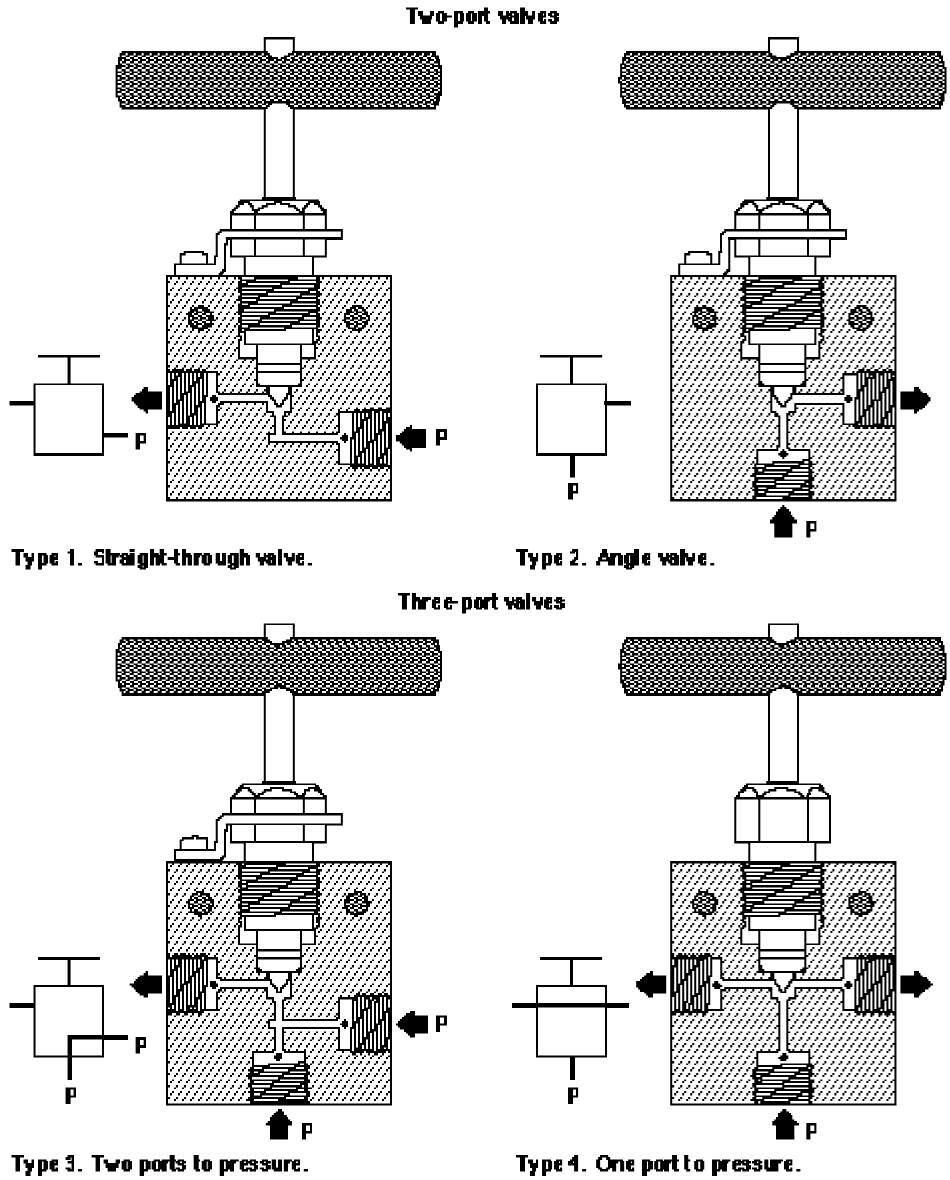
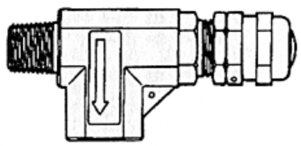


Figure 12. High-pressure valve body patterns.

Note: All pressure ports are marked "P"; nonpressure ports are unmarked. When valves are CLOSED, pressure ports are not exposed to valve-stem packing. Other ports are always exposed to valve-stem packing when the valve is OPEN or CLOSED.

6.6 Relief Devices

Pressure sources are to be limited to the MAWP of the lowest rated system component. When sources cannot be limited, the use of a pressure-relief device is required. Common relief devices (shown below) include a spring-loaded relief valve and a rupture disc assembly.

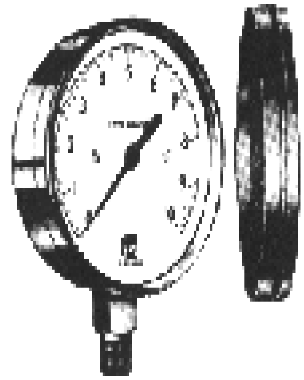


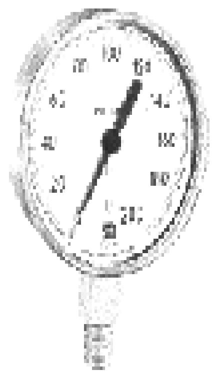
The following precautions apply to all pressure relief devices:

- Protect all manned-area pressure vessels by a relief device set at a pressure not exceeding the MAWP of the vessel.
- Whenever possible, use ASME code-approved (ASME UG-125-136) or specially stocked relief devices.
- Inspect, reset, or replace all relief devices on a periodic basis. A 3-year minimum interval is required. If an outside contractor installed the relief devices, have them rechecked at the end of the contract period.
- Never place a valve between a relief device and the component it is installed to protect.
- Never set a relief device above the MAWP of the lowest rated system component(s) it is installed to protect.
- Locate and orient relief devices so that their discharge is not hazardous to personnel.
- Install relief devices of adequate total flow capacity. When all supply ports are open, the pressure must never exceed 110% of the MAWP.
- Do not reset relief devices unless specifically authorized to do so. No LLNL worker is permitted to set, seal, or stamp relief devices on utility water boilers, steam boilers, and compressed-air receivers that are under the jurisdiction of the State of California. Only authorized workers in Bldgs. 511 and 875 (Maintenance) and Bldg. 343 (High-Pressure Test Facility) are permitted to set and seal relief devices on noncoded pressure vessels and systems.

6.7 Pressure Gauges

Pressure gauges (shown below) ts that indicate system pressure. These gauges are available with :ions, levels of accuracy, materials of construction, and pr





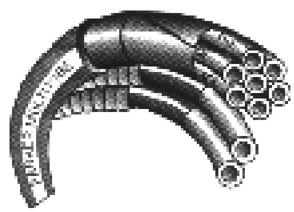
When selecting or installing a pressure gauge, consider the following:

- Use gauges graduated to about twice the MAWP of the system; never use gauges less than 1.2 times the MAWP. Be sure that gauge materials are compatible with the system fluid. (These rules apply to liquid as well as gas pressure gauges.)
- Use safety-type gauges (with shatter-proof faces, solid fronts, and blowout backs) or protect operators with a tested, approved gauge safety shield. This applies to all *gas* pressure gauges more than 4 inches (100 mm) in diameter and graduated to over 200 psig (1.4 MPa), *gas* pressure gauges less than 4 inches in diameter and graduated to over 5000 psig (34.5 MPa), and all *liquid* pressure gauges more than 4 inches in diameter and graduated to over 20,000 psig (138 MPa).
- Protect a gauge that is subject to excessive pressure surges or cyclic pulses by installing a throttling device, such as a pulsation dampener (preferred), a pressure snubber, a gauge saver, or a restricting orifice. Some gauges use a throttle screw in the tube socket to dampen surges.
- Make sure there is no oil or organic materials in gauges used on oxygen systems, because hydrocarbons and oxygen can combine explosively. Never use a gauge for oxygen that has been previously used on any other service. Clean all gauges used on high-purity gas systems.

- Protect gauges with a relief device to prevent the pressure from exceeding the full-scale reading of the gauge.
- Never use liquid-filled gauges with strong oxidizing agents such as oxygen, chlorine, or nitric acid.

6.8 Flexible Hose

Use a flexible hose (shown below) only where it is impractical to use metal tubing or pipe. Flexible hoses have a limited life, dependent on a given service, and failure to follow the manufacturer's recommended actions can result in a shortened service life or failure of the hose. The maximum recommended shelf life for rubber hose is approximately 8 years.



When specifying and installing a flexible hose, consider the following:

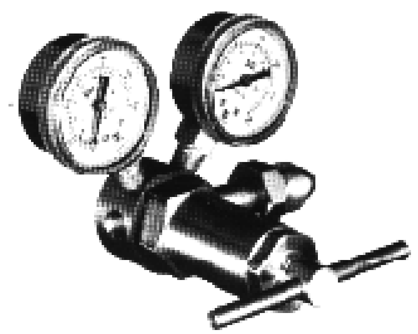
- **Rated working pressure.** Do not use flexible hoses at pressures over 1/4 of the minimum rated burst pressure stated by the manufacturer.
- **Fluid compatibility.** Do not use toxic or radioactive fluids since gases tend to permeate through hoses. Specially approved hoses may be used in certain flammable gas applications.
- **Sharp bends**. Do not bend or flex hose to a radius smaller than recommended; do not subject hose to torque or tension.
- **Hose ends.** Secure all hose ends with a hose restraint to prevent "whipping" in the event the hose or fitting fails. This requirement also applies where two hoses are coupled together.
- **Hose length and routing.** Keep the hose length as short as possible. Consider length changes under pressure, motion, and vibration. Protect or guide the hose to minimize abrasion, kinking, or excessive flexing.
- **Periodic inspection**. Have maintenance personnel perforate inert gas hose to prevent blistering. Repair or replace any hose showing leaks (pinholes), burns, wear blistering, or other defects.

6.9 Flash Arrestors and Check Valves

- Equip every flammable gas-drop regulator hose connection with a flash arrestor or check valve. If the flammable gas is to be (or could be) cross-connected with oxygen or compressed air, install a flash arrestor in the flammable gas line and place a check valve in the oxygen or compressed air line (see “Laboratory Gas Systems,” PEL-M-13200). This requirement applies to all single- and multiple-station installations and to all portable equipment.
- Equip all oxygen drops with a check valve. This requirement applies to single- and-multiple-station installations and portable equipment.

6.10 Regulators

The distribution systems for gas cylinders consist of a regulator (shown below) and a manifold. For a cylinder to be effective and safe, the regulator must take in gas from the cylinder and reduce the pressure to a low working pressure while simultaneously controlling the flow rate. It is important to obtain the correct regulator and ensure it is consistent with the gas involved and the operation intended. Manifolds distribute and control the gas flow from regulators.



The following precautions apply to all regulators:

- **Do not consider a pressure regulator by itself as satisfactory over pressure protection.**
- Never attempt to repair regulators. This shall only be done by authorized maintenance workers.
- Regulators should be taken to authorized personnel in the Plant Engineering Instrument Shop (Bldg. 511), the ME High-Pressure Test Facility (Bldg. 343), or to Site 300 (Bldg. 875) for inspection, adjustment, and tagging.
- For temporary storage, place used regulators in plastic bags to keep them clean.

- Survey work areas periodically for surplus regulators. Send all surplus regulators to authorized maintenance personnel for examination, cleaning, adjustment, repair, and tagging for future use.
- When removing regulators from flammable, toxic, or radioactive systems, make sure that all hazardous gas has been safely vented (and purged if required) from the entire regulator.
- Use only regulators that are designed and approved for the gas and cylinder with which they are used. Make sure that the Compressed Gas Association connection on the regulators corresponds with that on the cylinder valve outlet. Never force connections that do not fit. Make sure the cylinder valve and regulator connections are free of dirt, oil, grease, and any other foreign material. Use only oxygen regulators for oxygen service.
- Do not lubricate any part of the regulator or cylinder valve.
- Properly label regulators with the fluid being used.
- Only use line regulators up to a maximum pressure of 150 psig (1 MPa) for inline installations.
- Immediately replace damaged, defective, or unreliable regulators.

Single-stage cylinder regulators (except acetylene regulators) are equipped with a single relief device that is set to open at a value below the highest graduation on the low-side gauge. Authorized maintenance workers may also adjust these regulators to limit the output pressure to 75% of the highest output-gauge reading.

Two-stage regulators for inert gas are equipped with two relief valves that protect the regulator diaphragms and gauges from excessive overpressure. Relief valves on regulators for use with flammable, toxic, or radioactive gases must be safely vented.

At LLNL, two-stage regulators are adjusted so that the output pressure does not exceed 75% of the highest output-gauge reading. The low-side relief valve is set to open at a value below the highest graduation on the low-side gauge.

It is recommended that regulators be inspected every five years, but this is not a requirement.

6.11 Manifolds

6.11.1 Industrial Gas Cylinder Manifolds

Before submitting a job order for a manifold, make arrangements with the Supply and Distribution Department of the Industrial Gas Section to obtain the gas cylinder supply needed and the storage requirements.

The Laboratory's requirements for high-pressure manifolds is that only qualified LLNL craftsmen (LLNL pressure installers and inspectors) shall be responsible for these manifolds because of the high pressures involved. Therefore, all compressed-gas cylinder manifolds for both job-order work and purchase-order contract work shall be supplied, inspected, pressure tested, and tagged by these workers. An assembled manifold provided by LLNL can be installed as a unit by others (from "Laboratory Gas Systems," PEL-M-13200).

Do not leave manifold pigtails disconnected; insects can clog them. Insects in oxygen pigtails can cause spontaneous ignition, creating enough heat and overpressure to burst the pigtail, valve, or manifold. Either replace empty cylinders immediately, or have the excess pigtails and valves removed or capped to keep the system clean.

6.11.2 Safety Manifolds

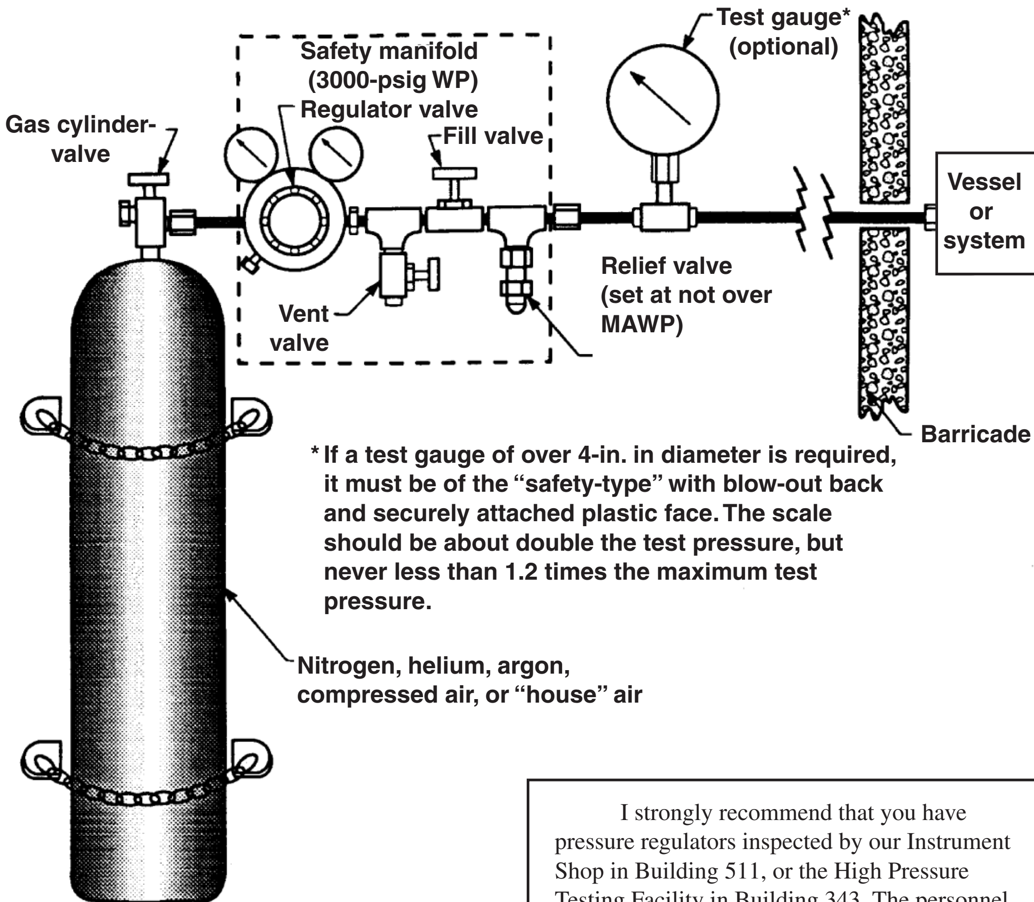
Authorized workers in Mechanical or Plant Engineering can provide safety manifold systems (see Fig. 13). These systems are designed to reduce the pressure from a standard cylinder and provide relief protection (relief device) for down-stream systems. Safety manifolds can be used for low-pressure (0–150 psi) applications that do not require formal documentation; at higher pressures, however, additional documentation (e.g., ESN or OSP) is required.

6.12 Temperature Considerations

Pressure hardware is usually rated at ambient temperature of 70˚F (21˚C). Sometimes manufacturers will designate an MAWP based on a lower or higher operating temperature. In general, the MAWP will decrease as the operating temperature increases. When selecting components, always ensure the fitting material as well as any seals and packing meet temperature requirements.

Following are temperature and working pressures for various sizes of copper-solder fittings:

Solder	Size (in)	Temperature (˚F)	Working pressure (psig)
95/5 (Tin-Antimony)	1/4 to 1	100	500
		150	400
		200	300
		250	200



I strongly recommend that you have pressure regulators inspected by our Instrument Shop in Building 511, or the High Pressure Testing Facility in Building 343. The personnel there will test this equipment for proper relief device settings, safety and reliability. They can also provide you with a safety manifold and a properly set relief device for your particular application.

Chuck Borzileri
Pressure Safety Manager
Extension 2-6076

Figure 13. Safety manifold system.

6.13 Installing Pressure Systems

All work on pressure equipment that requires an ESN must be done by or technically supervised by a certified LLNL pressure inspector, a pressure installer, or a closely supervised installer-in-training, under the direction of a responsible designer or responsible user.

6.14 Pressure Testing

Pressure test all systems in accordance with requirements in Document 18.3.

7.0 Work Standards

7.1 Work Smart Standards

8 CCR § 450-560, "Unfired Pressure Vessel Safety Orders (propane tanks, Air Receivers)."

29 CFR 1910.101, "Compressed Gases General Requirements."

29 CFR 1910.103, "Hydrogen."

29 CFR 1910.110, "Storage and Handling of Liquified Petroleum Gases."

29 CFR 1910.132, Subpart I, "Personal Protective Equipment."

29 CFR 1910.146, "Permit-required Confined Spaces."

29 CFR 1910, Subpart J, "General Environmental Controls."

49 CFR 100-199, "Hazardous Materials Transportation."

ANSI/B 31.1, "Power Piping Code."

ASME Boiler and Pressure Code, Section VIII, Division 1 "Rules for Construction of Pressure Vessels, and Division 2 "Alternative Rules" (latest version).

ACGIH TLVs and BEIs: Threshold Limit Values for Chemical Substances and Physical Agents, 2002 (excluding Biological Exposure Indices, TLVs for Physical Agents, and Biologically Derived Airborne Contaminants).

Compressed Gas Association (CGA), *Guidelines for handling of compressed gas cylinders. Pressure relief devices for large nonCode storage or process tanks*.

Compressed Gas Association, Pamphlet 1, "Safe Handling of Compressed Gases in Containers," 1991.

Compressed Gas Association, Pamphlet S-1.2, "Pressure Relief Device Standards" Part 2 *Cargo and Portable Tanks for Compressed Gases*, 1995.

Compressed Gas Association, Pamphlet S-1.3, "Pressure Relief Device Standards" Part 3 *Compressed Gas Storage Containers*, 1995.

Compressed Gas Asscoiation, Pamphlet P-12, "Safe Handling of Cryogen Liquids."

LLNL Pressure Safety Standard, Lawrence Livermore National Laboratory, Livermore, CA, UCRL-AR-128970.

NFPA 45, "Standard on Fire Protection for Laboratories Using Chemicals."

NFPA 51, "Standard for the Design and Installation of Oxygen-Fuel Gas Systems for Welding, Cutting, and Allied Processes."

NFPA 51B, “Standard for Fire Prevention in Use of Cutting and Welding Processes (1999).”

Public Law 91-596 § (5)(a)(1), OSHA General Duty Clause.

7.2 Other Required Standards

Plant Engineering Department, “Laboratory Gas Systems,” Lawrence Livermore National Laboratory, Livermore, CA, PEL-M-13200.

8.0 References

1.ASME Boiler and Pressure Vessel Code, Section VIII, “Pressure Vessels,” American Society of Mechanical Engineers, New York (latest version).

2.ASTM Standards, Vol. I, “Ferrous Metals,” American Society for Testing and Materials, Philadelphia, PA. (latest version).

3.*Strength of Metal Aircraft Elements*, Armed Forces Supply Support Center, Washington, D. C., Spec. MIL-HDBK-5 (latest version).

4.*Steel Forgings Alloy, High Yield Strength*, Bureau of Ships, Department of the Navy, Washington, D. C., Spec. Mil-S-23009 (April 1965).

5.*ME Design Safety Standards*, Lawrence Livermore National Laboratory, Livermore, CA, M-012 (latest version).

6.R. J. Roark, *Formulas for Stress and Strain*, (McGraw-Hill, New York, 1954), TS 265 R6 1954.

7.T. Baumeister, *Marks’ Mechanical Engineers’ Handbook* (McGraw-Hill, New York,1966), TJ 151 M486 1966.

8.F. B. Seely, *Advanced Mechanics of Materials*, (John Wiley and Sons, Inc., New York, 1952), 2nd ed.

9.F. Din, *Thermodynamic Functions of Gases*, (Butterworth Scientific Publications, London, 1956).

10. 49 CFR 100–199, “Research and Special Programs Administration.”

11. *Safety Standards for the Packaging of Fissile and Other Radioactive Materials*, Chapter 0529, U. S. Department of Energy, Washington, DC.

12. W. A. Burton, *ME Safety Note ENS 73-948*, “Gas Sampling Cylinders for LLL Shipments Containing Small Amounts of Radioactive Materials,” December 10, 1973.

13. ANSI / ASME B31.1, *Power Piping*, Parts 121.5, “Hanger Spacing,” p. 44 (1986 edition).

9.0 Resources for More Information

9.1 Contacts

For additional information about this document, contact the pressure safety

manager or the pressure consultant.

9.2 Lessons Learned

For lessons learned applicable to pressure vessels and systems, refer to the following Internet address:

http://www-r.llnl.gov/es_and_h/lessons/lessons.shtml

9.3 Other Sources

Air-Conditioning and Refrigeration Institute (ARI) Standards.

ASME Boiler and Pressure Vessel Code, Section VIII, "Pressure Vessels," Division 2, and Section X, "Fiber-Reinforced Plastic Pressure Vessels,"American Society of Mechanical Engineers, New York (latest version).

G. H. Bhat and D. V. Lindh, "Evaluation of Ultra-High Strength Steels for ThinWalled Pressure Vessels and Rocket Motor Cases," ASME Paper No. 62-MET-16 (1962).

R. Chuse, "Unfired Pressure Vessels," Nuclear Science Series TS-283-A (F. W. Dodge Corporation, New York, 1960).

E. W. Comings, *High Pressure Technology* (McGraw-Hill, New York, 1956).

J. P. Den Hartog, *Advanced Strength of Materials* (McGraw-Hill, New York, 1952), TA 405 D4, 1952.

T. J. Dolan, "Significance of Fatigue Data in Design of Pressure Vessels," *Welding J.* 35, 255s (1956) [ASME Paper No. 57-A-15 (1957)].

J. H. Faupel, *Engineering Design* (John Wiley and Sons, Inc., New York, 1964), TA 175 Fl, 1964.

G. Geroard, *Structural Significance of Ductility in Aerospace Pressure Vessels*, College of Engineering, New York, University, Tech. Rept. SM-60-8 (1960).

R. Gorcey, "Filament-Wound Pressure Vessels," *Design News*, Rocketdyne Division, North American Avation (January 1962).

J. F. Harvey, "Pressure Vessel Design: Nuclear and Chemical Applications," Nuclear Science Series TS-283-H2 (Van Nostrand Company, Princeton, N. J., 1963).

A. Hurlich and J. Balsch, "Titanium Pressure Vessels," *J. Metals*, **12**, 136 (1960).

G. R. Irwin, “Fracture of Pressure Vessels,” in *Materials for Missiles and Spacecraft*, E. R. Parker, Ed. (McGraw-Hill, New York, 1963), pp. 204–229.

N. L. Svensson, “The Bursting Pressure of Cylindrical and Spherical Vessels,” *ASME J. Appl. Mech.* **25**, 89 (1958).

H. Thielsch, *Defects and Failures in Pressure Vessels and Piping* (Reinhold Publishing Corporation, New York, 1965), TS 283 T3 1965.

S. Timoshenko, “Strength of Materials, Part 11.” *Advanced Theory and Problems* (Van Nostrand Company, Princeton, N. J., 1956).

D. A. Wruck, “Titanium Pressure Vessels,” *Machine Design*, **33**, 144 (1971).

Appendix A

Terms and Definitions

Brittle vessel	A pressure vessel fabricated from materials that do not yield extensively before failure when overstressed at any temperature within the specified working temperature range of the vessel. Materials that exhibit less than a 5% plastic strain to rupture are generally considered brittle.
Ductile vessel	A pressure vessel fabricated from materials that yield extensively before failure when overstressed at any temperature within the specified working temperature range of the vessel. Materials that exhibit greater than a 5% plastic strain to rupture are generally considered ductile. Some of these materials are listed in the ASME Boiler and Pressure Vessel Code, Section VIII (Division 1, Subsection C), and in Table 1 of this document.
Engineering Safety Note (ESN)	A management-approved (by division leader or higher) document that describes the anticipated hazards associated with a piece of equipment or a process. It describes the Responsible Individual's approach, analysis, and rationale used to assure the design safety of the equipment, system or process. An ESN does not have to be prepared by a member of the Engineering Directorate as long as the individual is technically qualified to prepare the ESN.
High pressure	Gas pressure greater than 3000 psig (20 MPa gauge), or liquid pressure greater than 5000 psig (35 MPa gauge).
Intermediate pressure	Gas pressure from 150 to 3000 psig (1–20 MPa gauge), or liquid pressure from 1500 to 5000 psig (10–35 MPa).
Low pressure	Gas pressure less than 150 psig (1 MPa), or liquid pressure less than 1500 psig (10 MPa).
Manned-area operation	Pressurization in environments where vessel failure might cause personal injury. Such vessels or systems have been approved for operation, within specified limits, with personnel present.

Manned-area vessel/ system	Pressurized vessels or systems approved for operation within specified limits and with personnel present.
Maximum allowable working pressure (MAWP)	The maximum pressure at which a vessel is designed to operate safely. Working pressure, rated pressure, service pressure, and design pressure are the same as MAWP. **Note**: The setting of vessel or system pressure-relief devices must not exceed this MAWP (see Fig. 2).
Maximum operating pressure (MOP)	The maximum pressure at which a pressure component is normally operated, usually 10–20% below the MAWP.
Pressure vessel	A relatively high-volume pressure component (such as a spherical or cylindrical container) that has a cross section larger than the associated pipe or tubing.
Remote operation	Pressurization in environments where vessel or system component failure would not cause personal injury. Remote operation equipment must be installed in test cells or behind certified barricades or must be operated from a safe location. Manned area operation (of a remote-operation vessel or system) for the purpose of leak checking or troubleshooting is limited to a maximum of 20% of the established and previously attained MAWP or 20% of the successful test pressure. Refer to Document 18.3, "Pressure Testing," in the *ES&H Manual* for pressure-testing and leak-checking requirements.
Safety factor	The ratio of the calculated failure pressure (or actual failure pressure if known) to the MAWP. A safety factor related to other than the failure pressure should be so identified with an appropriate subscript, i.e., SF_y for a safety factor based on the yield strength of the material, and SF_u for a safety factor based on its ultimate strength.

Appendix B

Example of an Engineering Safety Note

An engineering safety note is a management-approved (by division leader or higher) document that describes the anticipated hazards associated with a piece of equipment or a process. It describes the Responsible Individual's approach, analysis, and rationale used to assure the design safety of the equipment, system or process. An ESN does not have to be prepared by a member of the Engineering Directorate as long as the individual is technically qualified to prepare the ESN.

The new designations for ESNs are as follows:

- Mechanical Engineering Safety Note, MESN 99-001-OA
- Electronic Engineering Safety Note, EESN 99-001-OA
- Livermore Laboratory Safety Note, LLSN 99-001-OA

Assignment of a safety note number is controlled by the Engineering Records Center, Building 131, Room 1518.

http://www.llnl.gov/eng/MMED/home1.shtml

Mechanical Engineering Safety Note

500 PSIG Test Vessel

by

David C. Holten

October 1999

Approved by: ______________________

Responsible Individual

____

Pressure Consultant

____

Division Reviewer

____

Division Leader

Distribution

HPL Library
Engineering Records Center
Pressure Safety Manager
Responsible Individual
Pressure Consultant
Division Reviewer
Division Leader
System Users
Interested Parties

500 PSIG Test Vessel

A. Description

This Safety Note covers test vessel used to contain inert, liquid, or gas at pressures up to 500 psig. The vessel comprises a cylindrical section made of machined stainless steel pipe (8-inch nominal pipe size, schedule XXS) with a threaded/welded bottom and flanged/o-ring sealed lid. Three feed-throughs enter the lid via threaded pipe connections (2 ports are 1/2 inch pipe size; 1 port is 1/8 inch pipe size). LLNL Drawing Number AAA88-111390 (attached) describes this vessel.

B. Hazard

This vessel represents a potential hazard to personnel and equipment when pressurized to 500psi with liquid or gas. The latter case involves the greater stored energy and will therefore be calculated. The energy contained in the gas, assuming a reversible adiabatic (isentropic) expansion, is given by:

$$E = \frac{P_1 V_1}{K-1}\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{K-1}{K}}\right] \qquad \text{(Page C-11, Ref. [11])}$$

Where P_1 = MAWP
P_2 = Atmospheric pressure
V_1 = Volume of the vessel
K = C_p/C_v = Ratio of specific heats
E = Stored energy

For this design: P_1 = 500 psia
P_2 = 14.7 psia
V_1 = 11,305 cc
K = 1.4 for air or nitrogen (worst case)

Adjusting this equation for the proper units and substituting values gives:

$$E(\mathrm{gmTNT}) = 1.492\times10^{-6}\,\frac{P_1(\mathrm{psi})V_1(\mathrm{cc})}{K-1}\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{K-1}{K}}\right]$$

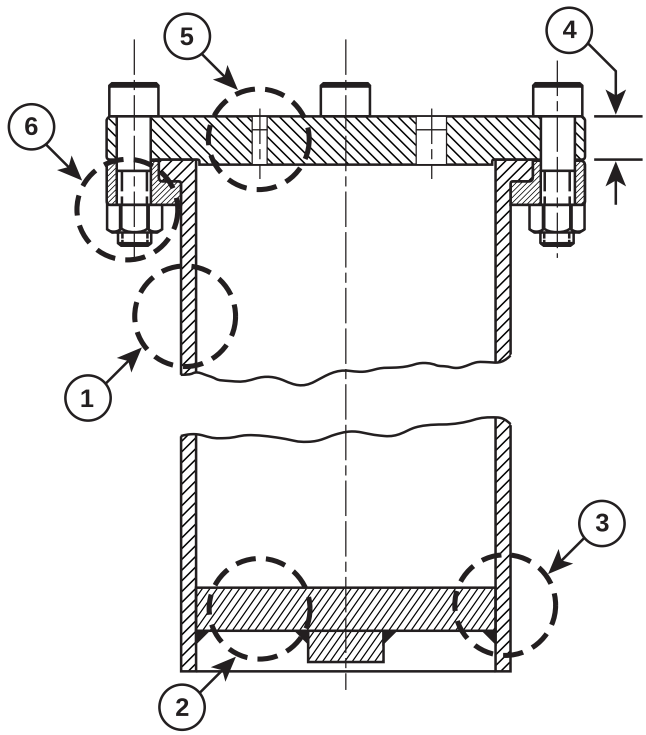
$$E(\mathrm{gmTNT}) = \frac{1.492\times10^{-6}\times500\times11{,}305}{1.4-1}\left[1-\left(\frac{14.7}{500}\right)^{\frac{1.4-1}{1.4}}\right]$$

E(gmTNT) = 13.4 (gmTNT)

C. Calculations

For this vessel, the following design features will be analyzed:

1. Cylindrical hoop stress
2. Bottom thickness
3. Bottom thread shear
4. Lid thickness
5. Lid feed-throughs
6. Bolt/nut thread stress



Design details for this vessel are as follows:

- 32°F to 130°F temperature operation
- Gasket seal = o-ring
- Vessel materials
  - Cylinder: 304 stainless steel per SA-479
  - Allowable stress: S_{a-v} = 18.800 psig (Ref. 2)

- Yield stress: Sy = 30,000 psig
- Allowable stress: $S_{a\text{-}v}$ = 18.800 psig (Ref. 3)

- Bolts

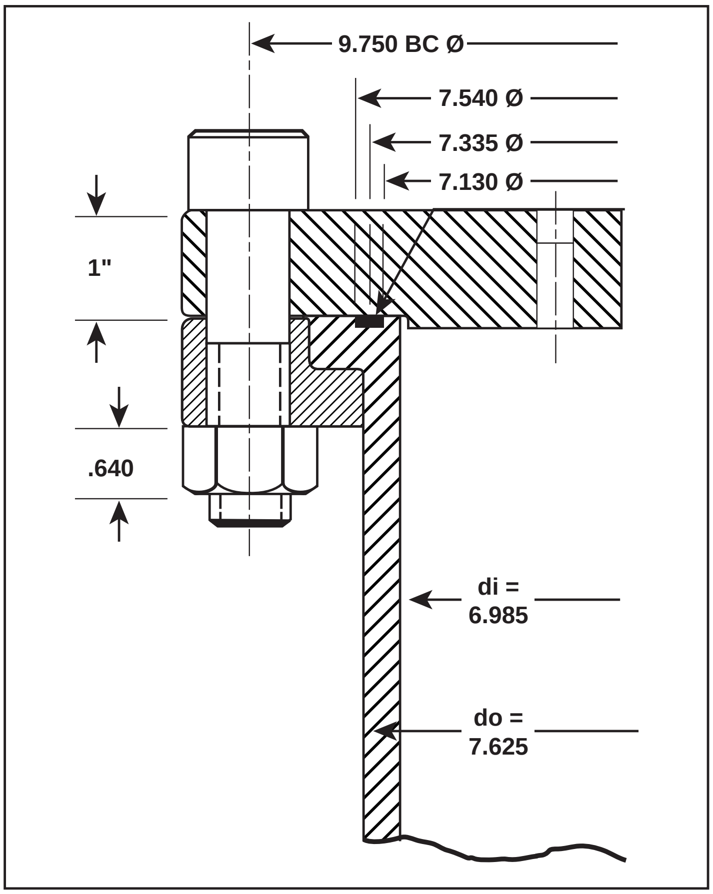
  8 each – 3/4-10UNC Soc. HD

  LLNL S/N 5305-20496

  S(ultimate = 160,000(psig) = $S_{u\text{-}b}$ (Ref. [11])

$$S(\text{allowable}) = \frac{S_{u-b}}{4} = 40,000(\text{psi}) = S_{a-b}$$

Detail of Vessel/Lid Seal



- Nuts

  LLNL S/N 5310-21810

  S(ultimate = 90,000(psig) = $S_{u\text{-}n}$ (Ref. [10])

$$S(\text{allowable}) = \frac{S_{u-n}}{4} = 22,500(\text{psi}) = S_{a-n}$$

Thickness of nut = T_n = .640

C.1 Cylindrical Hoop Stress

r_o = 7.625/2 = 3.8125

r_i = 6.925/2 = 3.4925

R = r_o/r_i = 3.8125/3.4925 = 1.092 . . . (use thin-wall equation)

$$P = \frac{S_{a-v}Et}{r_i}$$ (pg. E-8, Eq.[1]. Ref. [1])

Where t = .320 in.

E = joint efficiency = 1*

*This vessel contains no structural welds. In this design no strength credit is taken for the bottom seal weld. Rather, its only purpose is to provide a gas seal. The pressure load is fully taken by the 7–12 thread support.

Substituting values gives

$$P = \frac{18,800 \times .320}{3.4925} = 1,723(\text{psi})$$

Vessel (based on hoop stress) is good to 1,723(psig)

Vessel MAWP = 500 psig $\Rightarrow$ **OK**

C.2 Bottom Thickness

T = 1.0

$$T = d_i\sqrt{\frac{CP}{S_{a-v}E}}$$ (pg. E-10, Eq. [9], Ref. [1])

Where C = Attachment coefficient

= .75 (Case Q, pg. E-11, Ref. [1])

Solving for P gives

$$P = \frac{S_{a-v}}{C}\left(\frac{T}{d_i}\right)^2$$

Substituting values gives

$$P = \frac{18{,}000}{.75}\left(\frac{1}{6.985}\right)^2 = 516(\text{psi})$$

Vessel (based on hoop stress) is good to 516 (psig)

Vessel MAWP = 500 psig $\Rightarrow$ **OK**

C.3 Bottom Thread Shear

Bottom thread is 7-12-2A × 1'' long.

Thread form is

has = .02706 (pg. 62, Ref. [4])

Dm = Major diameter (external thread)

= 7.000

Dp = Pitch diameter

= 7,000 – 2 (.02706)

6.94558

Shear stress in the threads is given by

$$\sigma_s = \frac{F}{A_s} \qquad [1]$$

Where F = Force tending to shear the threads

= Design pressure □ Xs area of bottom

$$= P \times \frac{\pi}{4} D_m^2 \qquad [2]$$

A = Shear area of thread (pg 103, Ref. [5])

$$= \frac{\pi E L_e}{2} \qquad [3]$$

Where E = Minimum thread pitch diameter

$= D_p$ (from page 6)

$= 6.94558$

L_e = Length of engagement

= Bottom thickness

$= 1.0$

Limiting this shear stress to one-half the design stress, as

$$\sigma_s = \frac{\text{design stress}}{2} \qquad \text{(Part 1, pg. 5, Ref. [6])}$$

$$\sigma_s = \frac{S_{a-v}}{2} \qquad [4]$$

Combining equations 1, 2, 3, and 4, and solving for P gives

$$P = \frac{S_{a-\nu} E L_e}{D_m^2} \qquad [5]$$

Substituting values into equation 5 gives

$$P = \frac{18{,}800 \times 6.94558 \times 1}{7^2} = 2{,}665(\text{psi})$$

Threads in vessel bottom good to 2,665 (psig)
Vessel MAWP = 500 psig $\Rightarrow$ **OK**

C.4 Lid Thickness

Cited references are from ASME Boiler and pressure Vessel Code, Section VIII, Division 1, 1992, (Ref. 7) unless otherwise stated.

The vessel lid thickness is

$$T = d\sqrt{\frac{CP}{S_{a-\nu}E} + 1.9\frac{Wh_g}{S_{a-\nu}Ed^3}} \qquad \text{(Eq. [10], pg. E-10, Ref. [1])}$$

Where

C = attachment coefficient (Case K, pg. E-11, Ref. [1])

E = .3

E = 1 (pg E-13, Ref. [1])

h_g = Radial difference between bolt circle and pressure seal circle (flange moment arm)

= (9.75-7.335)/2 = 1.2075

d = pressure seal diameter = 7.335

W = flange design bolt load

Solving for P gives

$$P = \left(\frac{T^2}{d^2} - \frac{1.9Wh_g}{S_{a-v}d^3} \right) \frac{S_{a-v}}{C} \qquad [6]$$

C.4.1 Determination of W (Appendix 2, para. 2.5, pg. 312)

$$W = \text{largest of} \begin{cases} W_{m1} \text{ (operating conditions)} \\ W_{m2} \text{ (gasket sealing)} \end{cases}$$

W_{m1} = $H+H_p$ (Eq. [1], pg. 313)

Where Hp = ∅ for self-energizing seals (o-ring) (Para C3a, pg. 313)

H = hydrostatic end force

= $\frac{\pi}{4} G^2 P$ (Eq. [1], pg. 313)

G = d = 7.335

Substituting values gives

$$W_{m1} = \frac{\pi}{4} 7.335^2 \times 500 = 21{,}128 \text{(lbs)}$$

W_{m2} = 0 for self-energizing seals (o-rings) (Para. C3b, pg. 313)

W = W_{m1} = 21,128(lbs) (operating conditions)

C.4.2 Bolt

XS areas are determined as (pg. 310 and Para. C3d, pg. 313)

A_m = total required XS area of bolting

$$= \text{larger of} \begin{cases} A_{m1} & \text{(operating conditions)} \\ A_{m2} & \text{(gasket sealing)} \end{cases}$$

$$A_{m1} = \frac{W_{m1}}{S_b}$$

S_a = Allowable bolt stress at atmospheric temperature

S_b = Allowable bolt stress at operating temperature

$S_a = S_b = S_{a\text{-}b}$

Substituting values gives

$$A_{m1} = \frac{21{,}128}{40{,}000} = 0.5282(\text{in}^2)$$

$$A_{m2} = \frac{W_{m2}}{S_a} = 0 \qquad \text{(from above)}$$

$A_m = A_{m1} = 0.5282(\text{in}^2)$

A_b = actual total XS area of bolting

= $8 \times$ XS area of 3/4-10 bolt

= $8 \times .334$ (pg. 8-12. Ref. [8])

= $2.672(\text{in}^2)$

$A_b \geq A_m \Rightarrow$ **OK** (Para. C3d, pg. 313)

C.4.3 W for Gasket Seating

$$W = \frac{(A_m + A_b)S_{a-\nu}}{2} \qquad \text{(Eq. [4], pg. 319)}$$

The above equation applies to hard gaskets to protect them from over tightening and flange overloading. This equation does not apply to self-energizing seals (o-rings), as gasket seating loads are considered zero.

Thus,

$W = W_{m1} = 21{,}128$ (lbs).

Finally, values are substituted into Eq. 6, page 8, to determine lid thickness, as:

$$P = \left[\left(\frac{1}{7.335}\right)^2 - \frac{1.9 \times 21{,}128 \times 1.2075}{18{,}800 \times 7.335^3}\right]\frac{18{,}800}{.3} = 755(\text{psi})$$

Vessel lid is good to 755 (psig)

Vessel MAWP = 500 psig $\Rightarrow$ **OK**

C.5 Lid Feed-Throughs

Three male threaded pipe feed-throughs are installed in the vessel lid. These stainless steel fittings have pressure ratings in excess of 3,000 (psig). The corresponding internally threaded holes in the vessel lid will be evaluated for their pressure rating based on thread shear stress. A conservative, approximate analysis for a tapered pipe thread can be made by using the minimum pitch diameter, E_o, outside pipe diameter, D, hand tight engagement length, L, and the equations of paragraph C.3, on page 6.

Pipe Size	$E_o = E$	D	$L_1 = L_e$	
1/8	.36351	.405	.1615	(pg. 1363, Ref. [9])
1/2	.75843	.840	.320	

As in paragraph C.3, Eq. 5, page 7,

$$P = \frac{S_{a-v} E L_e}{D^2}$$

Substituting values gives

$$P(1/8\text{thd}) = \frac{18{,}800 \times .3635 \times .1615}{.405^2} = 6{,}729(\text{psi})$$

$$P(1/2\text{thd}) = \frac{18{,}800 \times .75843 \times .320}{.840^2} = 6{,}466(\text{psi})$$

Vessel feed-through connections are good to 6,466(psig)+

Vessel MAWP = 500 psig $\Rightarrow$ **OK**

C.6 Bolt/Nut Thread Stress

Eight each 3/4-10 UNC nuts and bolts fasten the vessel lid to the vessel. As in equation [5], page 7, the pressure-stress relationship for this thread system can be shown to be:

- Bolts

$$P = 8\left[\frac{S_{a-b}EL_e}{D^2}\right]$$

Where E = pitch diameter of the external thread (bolt)

= .6773 (Ref. [4])

L_e = thickness of the nut

= .640 (Ref. [10])

D = o-ring seal diameter

= 7.335 (Ref. pg. 4)

Substituting values gives

$$P = 8\left[\frac{40,000 \times .6773 \times .640}{7.335^2}\right] = 2,578(\text{psi})$$

Bolts are good to 2,578(psig).

Vessel MAWP = 500 psig $\Rightarrow$ **OK**

- Nuts

$$P = 8\left[\frac{S_{a-n}EL_e}{D^2}\right]$$

Where E = pitch diameter of the external thread (nut)

= .6850 (Ref. [4])

Substituting values gives:

$$P = 8\left[\frac{22,500 \times .6850 \times .640}{7.335^2}\right] = 1,467(\text{psi})$$

Bolts are good to 1,467(psig).

Vessel MAWP = 500 psig $\Rightarrow$ **OK**

D. Pressure Testing

The assembled vessel shall be pressure tested as follows:

1. Pressure test with helium to 1.5 times MAWP; i.e., $1.5 \times 500 = 750$(psig). Hold pressure a minimum of 30 minutes.
2. Leak check with helium to 1 times MAWP; i,e, 500(psig). Any leakage detectable with a mass spectrometer leak detector hand probe is unacceptable.

All tests shall be performed by a high-pressure technician and witnessed by an LLNL pressure inspector.

D.1 Maximum Energy of Distortion Analysis

To ensure that this vessel does not yield during this 1.5 times MAWP pressure test, the following "Maximum Energy of Distortion Analysis" calculation is performed.

(Reference pg. E-25, Ref [1])

Stress at test pressure = S_{vm} (combined Von Mises stress)

$$S_{vm} = \sqrt{\frac{1}{2}\left[(S_1 - S_2)^2 + (S_2 - S_3)^2 + (S_3 - S_1)^2\right]} \qquad [7]$$

$$\text{Define } Z = \left(\frac{r_o}{r_1}\right)^2 = \left(\frac{3.8125}{3.4925}\right)^2 = 1.192$$

$$S_1 = \frac{P}{Z - 1}$$

$$= \frac{750}{1.192 - 1}$$

$$= 3{,}906(\text{psig})$$

$$S_2 = P \times \left(\frac{Z + 1}{Z - 1}\right)$$

$$= 750\left(\frac{1.192 + 1}{1.192 - 1}\right)$$

$$= 8{,}562(\text{psig})$$

S_3 = -P

= -750(psig)

Substituting into equation [7] gives the following

S_{vm} = 8,064(psig)

N = Ratio of yield strength to combined stress

N ≥ 1 to ensure no yielding

S_y = 30,000(psig) (from pg. 3)

$$N = \frac{30,000}{8,064} = 3.72$$

Cylindrical vessel section will not yield during 1.5 times MAWP pressure test.

E. Re-test/Re-inspection

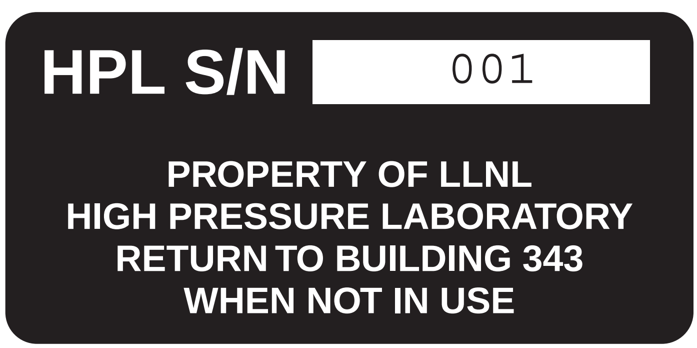
This system requires a re-inspection every three years and a re-test every six years. These shall be performed by an LLNL high-pressure technician and witnessed by an LLNL pressure inspector. Re-testing shall be done at 1 times the manned area MAWP previously defined in paragraph D.

F. Labeling

The pressure inspector will certify the inspection of this system by completion of an LLNL Pressure Test/Inspection Record, Form LL3586, and by attaching an LLNL Pressure Tested Label, filled out as follows:

G. Special Use Label

This item may be part of the High Pressure Testing Facility equipment inventory. In this category, it is maintained, utilized, and controlled by that facility. Such equipment may be made available, on a loan basis, to other LLNL projects. If in this category, the following label will be affixed:



H. Associated Documentation

1. AAA 88-111390 500(psig) test vessel
2. M.E. 1343 M.E. Test/Inspection Record for 500(psig) test vessel.

I. References

1. *DOE Pressure Safety Manual*, December 1993.
2. ASME Boiler and Pressure Vessel Code, Section 2, part D, subpart 1, table 1A, page 98.
3. ASME Boiler and Pressure Vessel Code, Section 2, part D, subpart 1, table 1A, page 98.
4. ASA B1.1 – 1960, Unified Screw Threads, ASME.
5. NBS 1963 Supplement to Screw-thread Standards for Federal Service, 1963 Supplement to H-28.
6. NBS Screw Thread Standards for Federal Service, Handbook H-28 (1957), part 1.
7. ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, 1992.
8. Marks Standard Handbook for Mechanical Engineers, 9th edition.
9. *Machinery Handbook*, 22nd edition.
10. LLNL ESR #354-5A.
11. LLNL ESR #283-1C.

Prepared by: Kaikinger, Wadleigh Date: March 17, 1975	**ENGINEERING STANDARD REFERENCE** **ISSUED BY**	LLL Stock Class 5305 ESR No. 283-1C Page 1 of 1
	LAWRENCE RADIATION LABORATORY **STANDARDS & SPECIFICATIONS GROUP,** **MECHANICAL ENGINEERING LIVERMORE**	SPECIFICATION FF-S-86D TYPE I MS-35455

TITLE: Screw, Cap, Socket Head, Hexagon; 1936 series, alloy, steel, UNC, class 3A thread, plain or knurled head, black oxide finish

APPLICATION: Where high strength and close tolerance threads are required.

TYPICAL DRAWING CALLOUT: Not Applicable

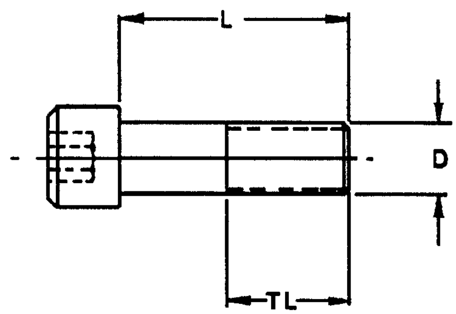
DO NOT USE FOR DESIGN. SEE ESR 285-5, 1960 SERIES. 1936 SERIES FOR REPLACEMENT ONLY IN COUNTER BORED HOLES. DESIGN, FABRICATION AND ASSEMBLY INFORMATION - see NBS H-28, ASA B1.1 or ESR 286 extract.

MATERIAL

type 4037, 4137, 4140, 8630, 8740
or equal which meets mechanical properties

CHEMICAL COMPOSITION (1)

Element	Percent Min	Percent Max
carbon	0.28	0.43
manganese	0.70	1.0
silicon	0.20	0.35
phosphorus	--	0.04
sulfur	--	0.04
chromium	--	1.10
nickel	--	0.07
molybdenum	0.15	0.30
iron	remainder	



(1) May vary to meet mechanical preterits

*MECHANICAL PROPERTIES

ultimate tensile strength min. psi 160,000
0.2 percent offset yield strength, min psi 130,000
Rockwell hardness C scale over 1/2 in. dia . . . . .36 to 43
percent reduction in are min 33
percent elongation min. 8
Rockwell hardness C scale 0 through 1/2 in. dia. . .36 to 45

*Specification requirements

LLL STOCK

LLL Stock No.	(D) Bolt Size	No. Threads per Inch T/Inch	(TL) Thread Length	(L) Length	MS-35455 (Dash)	Min. Tensile lbs	Min. Yield lbs	Key LLL Stock No. 5120-	Price	Vendor No.	Qty	Other
*5305-20461	5/16	- 18	1/2	1/2	47	8190	6660	26401				
*5305-20462	5/16	- 18	3/4	3/4	49	8190	6660	26401				
*5305-20463	5/16	- 18	1	1	51	8190	6660	26401				
*5305-20464	5/16	- 18	1 1/8	1 1/4	52	8190	6660	26401				
*5305-20465	5/16	- 18	1 1/8	1 1/2	53	8190	6660	26401				
*5305-20466	5/16	- 18	1 1/8	2	55	8190	6660	26401				
*5305-20487	5/8	- 11	1	1	118	35600	28900	27405				
*5305-20488	5/8	- 11	1 1/4	1 1/4	119	35600	28900	27405				
*5305-20489	5/8	- 11	1 1/2	1 1/2	120	35600	28900	27405				
*5305-20490	5/8	- 11	1 3/4	2	122	35600	28900	27405				
*5305-20491	5/8	- 11	1 3/4	2 1/2	124	35600	28900	27405				
*5305-20492	5/8	- 11	1 3/4	3	126	35600	28900	27405				
*5305-20493	3/4	- 10	1 1/2	1 1/2	135	52800	42900	27406				
*5305-20494	3/4	- 10	2	2	137	52800	42900	27406				
*5305-20495	3/4	- 10	2	2 1/2	139	52800	42900	27406				
*5305-20496	3/4	- 10	2	3	141	52800	42900	27406				
*5305-20497	3/4	- 10	2	3 1/2	143	52800	42900	27406				

* No Longer Stocked.

FOR LLL USE ONLY

Prepared by: Kaikinger, Wadleigh
Date: January 26, 1973

ENGINEERING STANDARD REFERENCE

ISSUED BY

LRL Stock Class 5310
ESR No. 354-5A
Page 1 of 2

LAWRENCE RADIATION LABORATORY

STANDARDS & SPECIFICATIONS GROUP,
MECHANICAL ENGINEERING LIVERMORE

SPECIFICATION FF-N-836D
SPECIFICATION MEL-503545

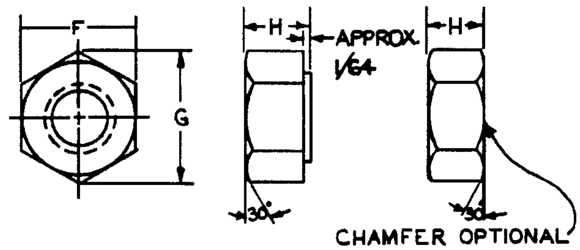
TITLE: Nut, Plain, Hexagon, steel, zinc, cad or nickel plated, UNC/UNF and NS class 2B thread
APPLICATION: Where fabrication requires wrench assembly with bolts and screws of equal strength.
TYPICAL DRAWING CALLOUT: SEE individual style.

CHEMICAL COMPOSITION (1)

Element	Percent Min	Percent Max
phosphorus	--	0.13
sulfur	--	0.04
carbon not controlled		

FINISH
zinc, cadmium or nickel

MECHANICAL PROPERTIES
tensile strength (proof load)
min psi . . . 90,000



STYLE 1: (finished) zinc/cadmium plated

TYPICAL DRAWING CALLOUT

Part No.	Description	Material	Specification	Stock No.
1/4-20	nut, plain, hexagon	low carbon steel zinc/cad finish	FF-N-836, grade A, type II, style 6	5310-21832

LRL STOCK

LLL Stock No.	UNC Thread	UNF Thread	N.S. Thread	Nom. inches Diameter Across Flat (F)	Max. inches Across Points (G)	Nom. inches Thick (H)	Min. Lb. Proof Load	Price	Vendor No.	Qty	Other
5310-21832	1/4-20			7/16	0.505	7/32	63180				
5310-21833		1/4-28		7/16	0.505	7/32	3280				
5310-21834	5/16-18			1/2	0.577	17/64	5240				
5310-21835		5/16-24		1/2	0.577	17/64	5220				
5310-21836	3/8-16			9/16	0.650	21/64	7750				
5310-21837		3/7-24		9/16	0.650	21/64	7900				
5310-21805		7/16-20		11/16	0.794	3/8	10680				
5310-21806	1/2-13			3/4	0.866	7/16	14200				
5310-21807		1/2-20		3/4	0.866	7/16	14390				
5310-21808	5/8-11			15/16	1.083	35/64	22600				
5310-21809		5/8-18		15/16	1.083	35/64	18270				
5310-21810	3/4-10			1 1/8	1.299	41/64	33400				
5310-21811		3/4-16		1 1/8	1.299	41/64	33510				
5310-21812	7/8-9			1 5/16	1.516	3/4	46200				
5310-21813		7/8-14		1 5/16	1.516	3/4	45810				
5310-21814	1-8			1 1/2	1.732	55/64	60600				
5310-21815			1-14*	1 1/2	1.732	55/64	--				
5310-21816	1 1/4-7			1 7/8	2.165	1 1/16	96900				
5310-21817	1 1/2-6			2 1/4	2.598	1 9/32	140500				

* formerly NF thread

STYLE 10: (Machine screw) zinc/cadmium plated (primary use in electronics)

TYPICAL DRAWING CALLOUT

Part No.	Description	Material	Specification	Stock No.
0-80	nut, plain, hexagon	low carbon steel zinc/cad finish	FF-N-836, grade A, type II, style 10	5310-21818

(LRL Stock continued on following page)

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Prepared by: Kaikinger, Wadleigh
Date: January 26, 1973

ENGINEERING STANDARD REFERENCE

LRL Stock Class 5310
ESR No. 354-5A

ISSUED BY

LAWRENCE RADIATION LABORATORY

STANDARDS & SPECIFICATIONS GROUP,
MECHANICAL ENGINEERING LIVERMORE

SPECIFICATION FF-N-836D
SPECIFICATION MEL-503545

LRL STOCK

LLL Stock No.	UNC Thread	UNF Thread	Nom. inches Diameter Across Flat (F)	Max. inches Across Points (G)	Nom. inches Thick (H)	Price	Vendor No.	Qty	Other
5310-21818		0-80	5/32	0.180	3/64				
5310-21819		1-72	5/32	0.180	3/64				
5310-21820	2-56		3/16	0.217	1/16				
5310-21821	3-48		3/16	0.217	1/16				
5310-21822		3-56	3/16	0.217	1/16				
5310-21823	4-40		1/4	0.289	3/32				
5310-21824	5-40		5/16	0.361	7/64				
5310-21825	6-32		5/16	0.361	7/64				
5310-21826	8-32		11/32	0.397	1/8				
5310-21827		8-36	11/32	0.397	1/8				
5310-21828	10-24		3/8	0.433	1/8				
5310-21829		10-32	3/8	0.433	1/8				
5310-21830	12-24		7/16	0.505	5/32				
5310-21831		12-28	7/16	0.505	5/32				

STYLE: Small pattern (machine screw) nickel plated

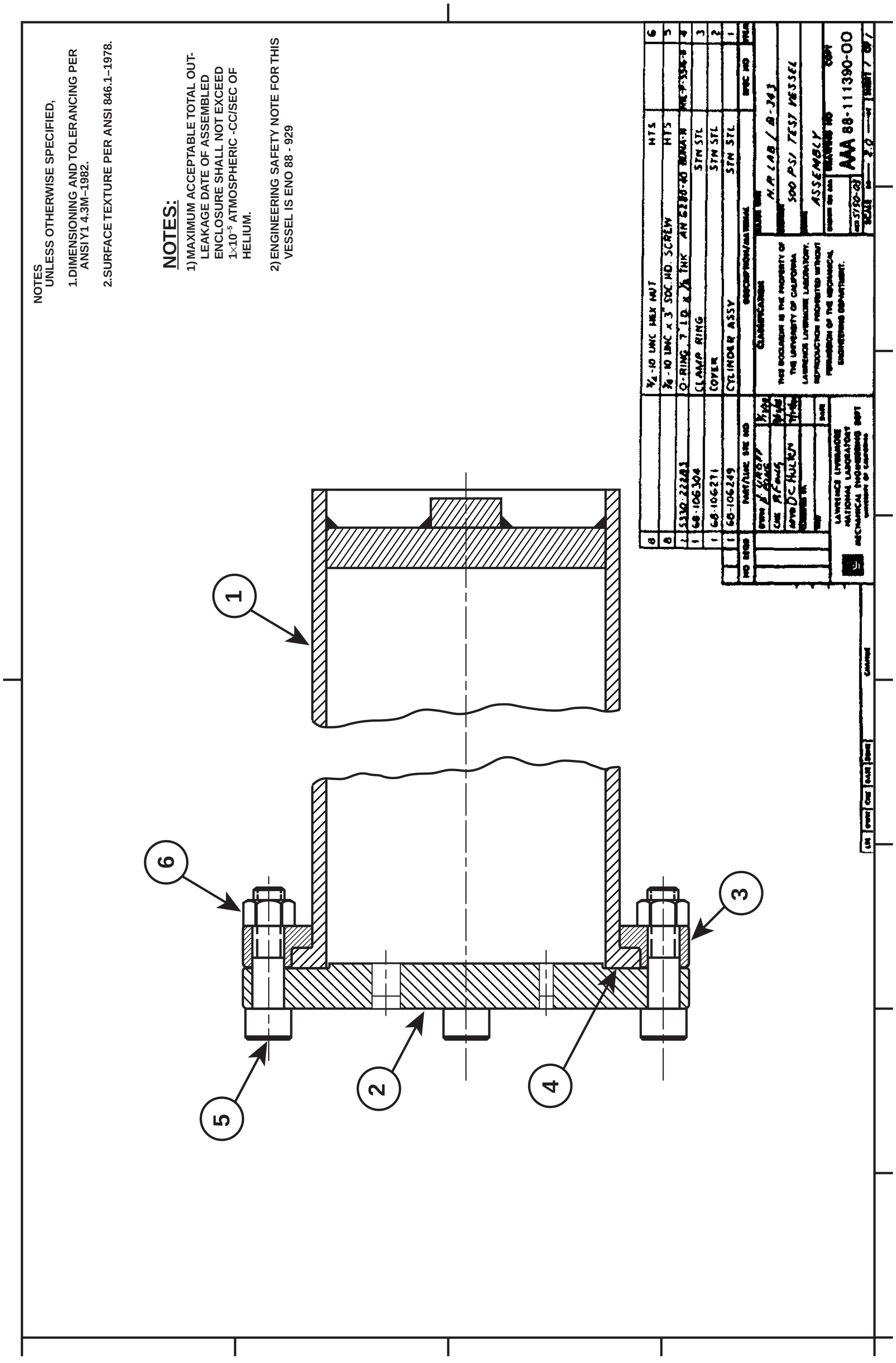
TYPICAL DRAWING CALLOUT

Part No.	Description	Material	Specification	Stock No.
6-32	nut, plain, hexagon	low carbon steel nickel finish	MEL-503845	5310-21838

LRL STOCK

LLL Stock No.	UNC Thread	UNF Thread	Nom. inches Diameter Across Flat (F)	Max. inches Across Points (G)	Nom. inches Thick (H)	Price	Vendor No.	Qty	Other
5310-21838	6-32		1/4	0.289	3/32				
5310-21839	6-32		1/4	0.289	3/32				

FOR LRL USE ONLY


NOTES
UNLESS OTHERWISE SPECIFIED,
1.DIMENSIONING AND TOLERANCING PER ANSIY1 4.3M–1982.
2.SURFACE TEXTURE PER ANSI 846.1–1978.
NOTES:
1) MAXIMUM ACCEPTABLE TOTAL OUT-LEAKAGE DATE OF ASSEMBLED ENCLOSURE SHALL NOT EXCEED 1×10⁻⁵ ATMOSPHERIC -CC/SEC OF HELIUM.
2) ENGINEERING SAFETY NOTE FOR THIS VESSEL IS ENO 88 - 929
1
2
3
4
5
6
CYLINDER ASSY
COVER
CLAMP RING
500 PSI TEST VESSEL
ASSEMBLY
AAA 88-111390-OO

Appendix C

ASME Pressure Vessel Code Guide

Table C-1 contains references to generally used information in the ASME Boiler and Pressure Vessel Codes, Section VIII, Division 1, "Pressure Vessels" (1992 edition):

Table C-1. References to general pressure vessel information.

Section	Subject	Page
US-23	Maximum allowable stress values (Subsection C starts on p. 141)	20
UG-27	Thickness of shells under internal pressure	23
UG-32	Formed heads, pressure on concave side	33
UG-34	Unstayed flat heads and covers (see Fig. UG-34, p. 4-48)	38
UG-101	Proof tests to establish Maximum Allowable Working Pressure	77
UG-125 to UG-136	Pressure relief devices	87-100
UW-9	Design of welded joints	107
UW-12	Joint efficiencies (see Table UW-12, p. 4-54)	109
Subsection C	Requirements pertaining to classes of materials (References Section II, Part D for maximum allowable stress values)	157
App. I	Supplementary design formulas (mandatory)	291
App. VI	Methods for magnetic particle examination (MT)	343
App. G	Suggested good practice regarding design of supports	477
App. L	Examples illustrating the application of code formulas and rules	481

Appendix D

Piping and Tubing Pressure Ratings

This appendix contains the maximum allowable working pressures (MAWP) for pipe and tubing generally available within the Department of Energy complex. The MAWP values shown were obtained using calculation methods and material properties taken from ANSI/B31.1, "Code for Power Piping (1989). A safety factor of 4 (5 for brass) is already calculated in these values.

The listed working pressures are for work temperatures up to 200°C (400°F) for ferrous pipe and tubing, and up to 120°C (250°F) for nonferrous pipe and tubing. These are given in both SI [megapascals (Mpa)] and English units [thousands of pounds per square inch (ksi)].

D.1 Threaded Pipe

Determine the MAWP of straight lengths of 1-in.-diameter, Schedule 40, UNS Alloy C23000, threaded brass pipe for a working temperature of 120°C. Use the following equation (see ANSI B31.1):

$$P = \frac{2\,SE\,(t_m - A)}{D_o - 2y(t_m - A)},$$

where

SE = Allowable stress at 120°C (250°F) [ANSI B31.1, App. A, pp. 182, 183] = 8.0 ksi.

t_m = Minimum wall thickness allowed under the specification = nominal wall thickness less wall tolerance = 0.126 – 0.007 = 0.119 in. (approximate value).

Note: To determine exact values, use the following.
For pipe t_m = 87.5% t_n (for brass pipe, use 94.5%).
For tubing t_m = 92.5% t_n (for stainless steel, use 87.5%).

A = Thread depth for 1-in.-o.d. pipe ["Dimension h," ANSI B2.1, Table 2, p. 7] = 80% of thread pitch; 1" pipe has 11.5 threads/in., so 80% (1/11.5) = 0.070 in.

D_o = Tabulated o.d. of 1-in. pipe = 1.315 in.

y = Coefficient for nonferrous pipe [ANSI B31.1] = 0.4. (Also for ferrous pipe.)

Substituting known values, the equation becomes

$$P = \frac{2 \times 8000(0.1190 - 070)}{1.315 - 2 \times 0.4(0.1190 - 070)} = 615 \text{ psig.}$$

In other units, this may be expressed as 0.61 ksi or 4.21 MPa.

D.2 Unthreaded Pipe

If the pipe in the previous example were assembled by brazing (A = 0.000), the MAWP would be 10.76 MPa (1.56 ksi).

$$P = \frac{2SE\,(t_m - A)}{D_o - 2y(t_m - A)},$$

or

$$P = \frac{2 \times 8000(0.1190 - 000)}{1.315 - 2 \times 0.4(0.1190 - 000)} = 1561 \text{ psig,}$$

or 1.56 ksi, or 10.76 MPa, which (in this case) is about 2.5 times its threaded rating.

D.3 Pipe Listing

Table No.	Type of piping
D-1.	Aluminum alloy, Schedule 40
D-2.	Brass, regular strength
D-3.	Brass, extra strength
D-4.	Copper, regular strength
D-5.	Copper, extra strength
D-6.	Black steel, Schedule 40
D-7.	Black steel, Schedule 80
D-8.	Carbon steel
D-9.	Stainless steel, Schedule 40

Aluminum Alloy Pipe

- Drawn, extruded, seamless.
- 6061-T6, Schedule 40 (UNS A96061).
- Per ASTM B241.
- Allowable stress: SE = 10500 psi (less than 1 in. IPS) or 9500 psi (1 in. or greater IPS).

Table D-1. Aluminum alloy, Schedule 40.

IPS	o.d.	Nominal wall thickness	MAWP			
			Threaded		Plain	
(in.)	(in.)	(in.)	(ksi)	(MPa)	(ksi)	(MPa)
1/4	0.540	0.088	1.34	9.24	3.38	23.31
3/4	1.050	0.113	0.86	5.93	2.14	14.76
1	1.315	0.133	0.68	4.69	1.80	12.41
1-1/2	1.900	0.145	0.58	4.00	1.34	9.24
2	2.375	0.154	0.53	3.66	1.13	7.79
3	3.500	0.216	0.49	3.38	1.07	7.38
5	5.563	0.258	0.43	2.97	0.79	5.45
6	6.625	0.288	0.42	2.90	0.72	4.97

Brass Pipe

- Seamless, annealed, regular strength.
- CDA Alloy 230 (UNS C23000).
- Per ASTM B43.
- Allowable stress: SE = 8000 psi.

Table D-2. Brass, regular strength.

IPS	o.d.	Nominal wall thickness	MAWP			
			Threaded		Plain	
(in.)	(in.)	(in.)	(ksi)	(MPa)	(ksi)	(MPa)
1/8	0.405	0.062	1.17	8.07	2.58	17.79
1/4	0.540	0.082	1.02	7.03	2.57	17.72
3/8	0.675	0.090	1.02	7.03	2.24	15.45
1/2	0.840	0.107	0.87	6.00	2.12	14.62
3/4	1.050	0.114	0.80	5.52	1.79	12.34
1	1.315	0.126	0.61	4.21	1.56	10.76
1-1/4	1.660	0.146	0.67	4.62	1.42	9.79
1-1/2	1.900	0.150	0.62	4.28	1.27	8.76
2	2.375	0.156	0.53	3.66	1.04	7.17
2-1/2	2.875	0.187	0.43	2.97	1.03	7.10
3	3.500	0.219	0.50	3.45	0.99	6.83
3-1/2	4.000	0.250	0.56	3.86	0.99	6.83
4	4.500	0.250	0.49	3.38	0.87	6.00
5	5.562	0.250	0.39	2.69	0.70	4.83
6	6.625	0.250	0.33	2.28	0.58	4.00

Brass Pipe

- Seamless, annealed, extra strength.
- CDA Alloy 230 (UNS C23000).
- Per ASTM B43.
- Allowable stress: SE = 8000 psi.

Table D-3. Brass, extra strength.

IPS	o.d.	Nominal wall thickness	MAWP			
			Threaded		Plain	
(in.)	(in.)	(in.)	(ksi)	(MPa)	(ksi)	(MPa)
1/4	0.540	0.123	2.38	16.41	4.15	28.62
1	1.315	0.182	1.32	9.10	2.33	16.07
2	2.375	0.221	0.98	6.76	1.51	10.41
3	3.500	0.304	0.89	6.14	1.40	9.66

Copper Pipe

- Seamless, drawn, regular strength.
- UNS Alloy C12200.
- Per ASTM B42.
- Allowable stress: SE = 11300 psi (1/8 to 2 in. IPS), 9000 psi (2-1/2 to 12 in. IPS).

Table D-4. Copper, regular strength.

IPS	o.d.	Nominal wall thickness	MAWP			
			Threaded		Plain	
(in.)	(in.)	(in.)	(ksi)	(MPa)	(ksi)	(MPa)
1/8	0.405	0.062	1.64	11.31	3.63	25.03
1/4	0.540	0.082	1.44	9.93	3.62	24.97
3/8	0.675	0.090	1.43	9.86	3.15	21.72
1/2	0.840	0.107	1.23	8.48	2.99	20.62
3/4	1.050	0.114	1.13	7.79	2.52	17.38
1	1.315	0.126	0.86	5.93	2.19	15.10
1-1/4	1.660	0.146	0.95	6.55	2.00	13.79
1-1/2	1.900	0.150	0.87	6.00	1.78	12.28
2	2.375	0.156	0.74	5.10	1.46	10.07
2-1/2	2.875	0.187	0.49	3.38	1.16	8.00
3	3.500	0.219	0.56	3.86	1.11	7.66
3-1/2	4.000	0.250	0.63	4.34	1.12	7.72
4	4.500	0.250	0.55	3.79	0.98	6.76
5	5.562	0.250	0.44	3.03	0.79	5.45
6	6.625	0.250	0.37	2.55	0.66	4.55

Copper Pipe

- Seamless, drawn, extra strength.
- UNS Alloy C12200.
- Per ASTM B42.
- Allowable stress: SE = 11300 psi (1/8" to 2" IPS), 9000 psi (2-1/2" to 12" IPS).

Table D-5. Copper, extra strength.

IPS	o.d.	Nominal wall thickness	MAWP			
			Threaded		Plain	
(in.)	(in.)	(in.)	(ksi)	(MPa)	(ksi)	(MPa)
1/2	0.840	0.149	2.44	16.83	4.36	30.07
3/4	1.050	0.157	2.09	14.41	3.57	24.62
1	1.315	0.182	1.86	12.83	3.28	22.62
1-1/4	1.660	0.194	1.63	11.24	2.73	18.83
1-1/2	1.990	0.203	1.52	10.48	2.47	17.03
2	2.375	0.221	1.38	9.52	2.13	14.69
2-1/2	2.875	0.280	1.08	7.45	1.79	12.34
3	3.500	0.304	1.01	6.97	1.58	10.90
4	4.500	0.341	0.92	6.34	1.37	9.45

Black Steel Pipe

- Seamless, Schedule 40.
- Carbon steel.
- Per ASTM A53, Grade B.
- Allowable stress: SE = 15000 psi.

Table D-6. Black steel, Schedule 40.

IPS	o.d.	Nominal wall thickness	MAWP			
			Threaded		Plain	
(in.)	(in.)	(in.)	(ksi)	(MPa)	(ksi)	(MPa)
1/8	0.405	0.068	2.27	15.66	4.94	34.07
1/4	0.540	0.088	1.92	13.24	4.82	33.24
3/8	0.675	0.091	1.67	11.52	3.92	27.03
1/2	0.840	0.109	1.40	9.66	3.73	25.72
3/4	1.050	0.113	1.24	8.55	3.05	21.03
1	1.315	0.133	1.08	7.45	2.84	19.59
1-1/4	1.660	0.140	0.96	6.62	2.34	16.14
1-1/2	1.900	0.145	0.92	6.34	2.11	14.55
2	2.375	0.154	0.83	5.72	1.78	12.28
2-1/2	2.875	0.203	0.83	5.72	1.95	13.45
3	3.500	0.216	0.77	5.31	1.69	11.66
3-1/2	4.000	0.226	0.75	5.17	1.54	10.62
4	4.500	0.237	0.72	4.97	1.43	9.86
5	5.563	0.258	0.69	4.76	1.26	8.69
6	6.625	0.280	0.66	4.55	1.14	7.86

Black Steel Pipe

- Seamless, Schedule 80.
- Carbon steel.
- Per ASTM A53, Grade B.
- Allowable stress: SE = 15000 psi.

Table D-7. Black steel, Schedule 80.

IPS	o.d.	Nominal wall thickness	MAWP			
			Threaded		Plain	
(in.)	(in.)	(in.)	(ksi)	(MPa)	(ksi)	(MPa)
1/8	0.405	0.095	4.38	30.21	7.35	50.69
1/4	0.540	0.119	3.65	25.17	6.83	47.10
3/8	0.675	0.126	3.18	21.93	5.62	38.76
1/2	0.840	0.147	2.76	19.03	5.25	36.21
3/4	1.050	0.154	2.36	16.28	4.29	29.59
1	1.315	0.179	2.09	14.41	3.96	27.31
1-1/4	1.660	0.191	1.83	12.62	3.28	22.62
1-1/2	1.900	0.200	1.73	11.93	2.98	20.55
2	2.375	0.218	1.59	10.97	2.57	17.72
2-1/2	2.875	0.276	1.53	10.55	2.69	18.55
3	3.500	0.300	1.44	9.93	2.38	16.41
3-1/2	4.000	0.318	1.38	9.52	2.20	15.17
4	4.500	0.337	1.34	9.24	2.07	14.28
5	5.563	0.375	1.27	8.76	1.85	12.76
6	6.625	0.432	1.30	8.98	1.79	12.37

Carbon Steel Pipe

- Seamless, cold drawn, stress relieved.
- Carbon steel, AISI MT-1016 1018.
- Per ASTM A106, Grade C.
- Allowable stress: SE = 17500 psi.

Table D-8. Carbon steel.

IPS	o.d.	Nominal wall thickness	MAWP*			
			Threaded		Plain	
(in.)	(in.)	(in.)	(ksi)	(MPa)	(ksi)	(MPa)
1/8	0.405	0.068	2.71	18.67	5.83	40.18
1/4	0.540	0.088	2.25	15.51	5.63	38.84
3/8	0.675	0.091	1.93	13.30	4.56	31.43
1/2	0.840	0.109	1.66	11.44	4.37	30.14
3/4	1.050	0.113	1.44	9.94	3.56	24.58
1	1.315	0.133	1.27	8.76	3.33	22.98
1-1/4	1.660	0.140	1.14	7.83	2.74	18.93
1-1/2	1.900	0.145	1.07	7.40	2.47	17.02
2	2.375	0.154	0.98	6.73	2.08	14.34
2-1/2	2.875	0.203	0.97	6.66	2.27	15.68
3	3.500	0.216	0.91	6.26	1.98	13.62
3-1/2	4.000	0.226	0.87	6.01	1.80	12.42
4	4.500	0.237	0.85	5.87	1.67	11.55
5	5.563	0.258	0.81	5.56	1.47	10.12
6	6.625	0.280	0.78	5.38	1.33	9.20

* Based on SAE J 1397, MT-1016.

Stainless Steel Pipe

- Seamless, annealed, Schedule 40.
- CRES, Type 304.
- Per ASTM A312.
- Allowable stress: SE = 18800 psi.

Table D-9. Stainless steel, Schedule 40.

IPS	o.d.	Nominal wall thickness	MAWP			
			Threaded		Plain	
(in.)	(in.)	(in.)	(ksi)	(MPa)	(ksi)	(MPa)
1/8	0.405	0.068	2.84	19.59	6.18	42.62
1/4	0.540	0.088	2.40	16.55	6.03	41.59
3/8	0.675	0.091	2.08	14.34	4.91	33.86
1/2	0.840	0.109	1.76	12.14	4.66	32.14
3/4	1.050	0.113	1.55	10.69	3.82	26.34
1	1.315	0.133	1.35	9.31	3.55	24.48
1-1/4	1.660	0.140	1.20	8.28	2.92	20.14
1-1/2	1.900	0.145	1.15	7.93	2.64	18.21
2	2.375	0.154	1.04	7.17	2.23	15.38
2-1/2	2.875	0.203	1.04	7.17	2.44	16.83
3	3.500	0.216	0.97	6.69	2.11	14.55
3-1/2	4.000	0.226	0.93	6.41	1.93	13.31
4	4.500	0.237	0.90	6.21	1.79	12.34
5	5.563	0.258	0.86	5.93	1.57	10.83

D.4 Tube Listing

Table No.	Type of tubing
D-10.	Aluminum alloy
D-11.	Brass, hard drawn
D-12.	Copper, H58 drawn, 12-ft lengths
D-13.	Copper, annealed, 50-ft coils, general use
D-14.	Copper, annealed, 50-ft coils, refrigeration use
D-15.	Copper, drawn temper, 20-ft lengths
D-16.	Copper, Type K
D-17.	Copper, Type L
D-18.	Copper, Type M
D-19.	Carbon steel
D-20.	Stainless steel

Aluminum Alloy Tubing

- Seamless, drawn.
- 6061-T6 alloy (UNS A96061).
- Per ASTM B210.
- Allowable stress: SE = 10000 psi.

Table D-10. Aluminum alloy.

o.d.	Nominal wall thickness	MAWP		o.d.	Nominal wall thickness	MAWP	
(in.)	(in.)	(ksi)	(MPa)				
0.375	0.035	1.79	12.34	1.000	0.065	1.21	8.34
0.375	0.049	2.58	17.79	1.000	0.083	1.60	11.03
0.375	0.058	3.09	21.31	1.125	0.058	0.94	6.48
0.500	0.035	1.32	9.10	1.250	0.049	0.72	4.97
0.500	0.049	1.88	12.97	1.250	0.065	0.96	6.62
0.500	0.065	2.56	17.66	1.250	0.083	1.26	8.69
0.625	0.049	1.48	10.21	1.500	0.065	0.79	5.45
0.625	0.065	2.01	13.86	1.500	0.083	1.04	7.17
0.750	0.035	0.86	5.93	1.625	0.058	0.64	4.41
0.750	0.049	1.22	8.41	2.000	0.049	0.44	3.03
0.750	0.058	1.45	10.00	2.000	0.065	0.59	4.07
0.750	0.065	1.65	11.38	2.500	0.065	0.47	3.24
0.750	0.083	2.18	15.03	3.500	0.065	0.33	2.28
0.875	0.065	1.40	9.66	—	—	—	—

Brass Tubing

- Seamless, hard drawn.
- CDA Alloy 280.
- Per ASTM B111.
- Allowable stress: SE = 10000 psi.

Table D-11. Brass, hard drawn.

o.d.	Nominal wall thickness	MAWP*		o.d.	Nominal wall thickness	MAWP*	
(in.)	(in.)	(ksi)	(MPa)	(in.)	(in.)	(ksi)	(MPa)
0.125	0.035	6.53	45.05	2.000	0.125	1.21	8.36
0.188	0.022	2.38	16.39	2.125	0.125	1.14	7.84
0.188	0.035	4.01	27.63	2.250	0.065	0.55	3.77
0.188	0.049	5.99	41.33	2.250	0.125	1.07	7.39
0.250	0.035	2.89	19.92	2.500	0.035	0.26	1.80
0.250	0.065	5.96	41.07	2.500	0.065	0.49	3.38
0.313	0.035	2.26	15.58	2.500	0.125	0.96	6.62
0.375	0.035	1.85	12.79	2.625	0.125	0.91	6.30
0.375	0.065	3.68	25.36	2.750	0.065	0.45	3.07
0.500	0.035	1.37	9.42	2.750	0.125	0.87	6.00
0.500	0.065	2.66	18.35	2.875	0.065	0.43	2.93
0.625	0.035	1.08	7.45	3.000	0.035	0.22	1.50
0.625	0.065	2.08	14.37	3.000	0.065	0.41	2.81
0.625	0.125	4.34	29.94	3.000	0.125	0.80	5.48
0.750	0.035	0.89	6.17	3.000	0.187	1.21	8.34
0.750	0.065	1.71	11.81	3.250	0.065	0.38	2.59
0.750	0.125	3.52	24.25	3.250	0.125	0.73	5.05
0.875	0.035	0.76	5.26	3.500	0.065	0.35	2.40
0.875	0.065	1.45	10.03	3.500	0.125	0.68	4.68
0.875	0.083	1.89	13.01	3.750	0.125	0.63	4.36
1.000	0.035	0.66	4.58	4.000	0.065	0.30	2.10
1.000	0.065	1.26	8.71	4.000	0.125	0.59	4.08
1.000	0.125	2.55	17.57	4.250	0.125	0.56	3.84
1.125	0.035	0.59	4.06	4.500	0.065	0.27	1.86
1.250	0.035	0.53	3.65	4.500	0.125	0.52	3.62
1.250	0.065	1.00	6.90	4.750	0.125	0.50	3.42
1.250	0.125	2.00	13.78	5.000	0.065	0.24	1.67
1.375	0.065	0.91	6.25	5.000	0.125	0.47	3.25
1.375	0.125	1.80	12.43	5.250	0.125	0.45	3.09
1.500	0.035	0.44	3.03	5.500	0.125	0.43	2.95
1.500	0.065	0.83	5.71	6.000	0.125	0.39	2.70
1.500	0.125	1.64	11.33	6.500	0.125	0.36	2.49
1.750	0.035	0.38	2.59	6.750	0.125	0.35	2.39
1.750	0.065	0.71	4.87	7.000	0.125	0.33	2.31
1.750	0.125	1.40	9.62	7.250	0.125	0.32	2.23
2.000	0.035	0.33	2.26	8.250	0.125	0.28	1.95
2.000	0.065	0.62	4.25				

* Based on MIL-T-46072.

Copper Tubing

- Seamless, H58 drawn, 12-ft straight lengths.
- Copper, UNS C12200, general use.
- Per ASTM B75.
- Allowable stress: SE = 9000 psi.

Table D-12. Copper, H58 drawn, 12-ft lengths.

o.d.	Nominal wall thickness	MAWP		o.d.	Nominal wall thickness	MAWP	
(in.)	(in.)	(ksi)	(MPa)	(in.)	(in.)	(ksi)	(MPa)
0.125	0.020	2.74	18.90	1.000	0.035	0.58	4.00
0.125	0.032	5.12	35.31	1.000	0.065	1.15	7.93
0.188	0.028	2.73	18.83	1.000	0.129	2.45	16.90
0.188	0.032	3.23	22.28	1.250	0.065	0.91	6.28
0.250	0.035	2.56	17.66	1.500	0.065	0.75	5.17
0.250	0.049	3.88	26.76	1.750	0.065	0.64	4.41
0.250	0.065	5.51	38.00	2.000	0.065	0.56	3.86
0.313	0.035	2.00	13.79	2.000	0.083	0.72	4.97
0.375	0.035	1.64	11.31	2.500	0.065	0.43	2.97
0.375	0.065	3.39	23.38	2.500	0.083	0.56	3.86
0.500	0.035	1.21	8.34	2.500	0.125	0.87	6.00
0.500	0.065	2.45	16.90	3.000	0.083	0.46	3.17
0.625	0.035	0.96	6.62	3.000	0.120	0.69	4.76
0.625	0.065	1.92	13.24	3.500	0.065	0.30	2.07
0.625	0.083	2.53	17.45	4.000	0.125	0.53	3.66
0.750	0.035	0.78	5.38	5.000	0.250	0.88	6.07
0.750	0.065	1.56	10.76	6.000	0.250	0.73	5.03
0.750	0.083	2.04	14.07	—	—	—	—

Copper Tubing

- Seamless, 060 soft annealed, 50-ft coils.
- Copper, UNS C12200, general use.
- Per ASTM B75.
- Allowable stress: SE = 6000 psi.

Table D-13. Copper, annealed, 50-ft coils, general use.

o.d	Nominal wall thickness	MAWP	
(in.)	(in.)	(ksi)	(MPa)
0.188	0.032	2.15	14.83
0.250	0.049	2.58	17.79
0.375	0.049	1.63	11.24
0.375	0.065	2.26	15.59
0.500	0.032	0.73	5.03
0.500	0.065	1.63	11.24

Copper Tubing

- Seamless, soft annealed, 50-ft coils.
- Copper, UNS C12200, refrigeration use.
- Per ASTM B280.
- Allowable stress: SE = 6000 psi.

Table D-14. Copper, annealed, 50-ft coils, refrigeration use.

o.d	Nominal wall thickness	MAWP		o.d	Nominal wall thickness	MAWP	
(in.)	(in.)	(ksi)	(MPa)	(in.)	(in.)	(ksi)	(MPa)
0.125	0.030	3.13	21.59	0.750	0.035	0.52	3.59
0.187	0.030	2.00	13.79	0.750	0.042	0.64	4.41
0.250	0.030	1.44	9.93	0.875	0.045	0.58	4.00
0.312	0.032	1.22	8.41	1.125	0.050	0.50	3.45
0.375	0.032	1.00	6.90	1.375	0.055	0.45	3.10
0.500	0.032	0.74	5.10	1.625	0.060	0.42	2.90
0.625	0.035	0.64	4.41				

Copper Tubing

- Seamless, drawn temper, 20-ft straight lengths.
- Copper, UNS C12200, air conditioning use.
- Per ASTM B280.
- Allowable stress: SE = 9000 psi.

Table D-15. Copper, drawn temper, 20-ft lengths.

o.d	Nominal wall thickness	MAWP		o.d	Nominal wall thickness	MAWP	
(in.)	(in.)	(ksi)	(MPa)	(in.)	(in.)	(ksi)	(MPa)
0.375	0.030	1.34	9.24	1.625	0.060	0.63	4.34
0.500	0.035	1.19	8.21	2.125	0.070	0.55	3.79
0.625	0.040	1.10	7.59	2.625	0.080	0.51	3.52
0.750	0.042	0.96	6.62	3.125	0.090	0.48	3.31
0.875	0.045	0.87	6.00	3.625	0.100	0.47	3.24
1.125	0.050	0.76	5.24	4.125	0.110	0.45	3.10
1.375	0.055	0.68	4.69				

Copper Tubing

- Seamless, hard drawn, 20-ft straight lengths.
- Copper, UNS C12200, Type K, plumbing use.
- Per ASTM B88.
- Allowable stress: SE = 6000 psi.*

Table D-16. Copper, Type K.

Water tube size	o.d	Nominal wall thickness	MAWP		Water tube size	o.d	Nominal wall thickness	MAWP	
(in.)	(in.)	(in.)	(ksi)	(MPa)	(in.)	(in.)	(in.)	(ksi)	(MPa)
1/4	0.375	0.035	1.06	7.31	2	2.125	0.083	0.44	3.03
3/8	0.500	0.049	1.16	8.00	2-1/2	2.625	0.095	0.41	2.83
1/2	0.625	0.049	0.91	6.28	3	3.125	0.109	0.40	2.76
5/8	0.750	0.049	0.75	5.17	3-1/2	3.625	0.120	0.38	2.62
3/4	0.875	0.065	0.89	6.14	4	4.125	0.134	0.37	2.55
1	1.125	0.065	0.68	4.69	5	5.125	0.160	0.36	2.48
1-1/4	1.375	0.065	0.55	3.79	6	6.125	0.192	0.36	2.48
1-1/2	1.625	0.072	0.51	3.52	8	8.125	0.271	0.38	2.62

* Annealed properties assumed.

Copper Tubing

- Seamless, hard drawn, 20-ft straight lengths.
- Copper, UNS C12200, Type L, plumbing use.
- Per ASTM B88.
- Allowable stress: SE = 6000 psi.*

Table D-17. Copper, Type L.

Water tube size	o.d	Nominal wall thickness	MAWP*		Water tube size	o.d	Nominal wall thickness	MAWP*	
(in.)	(in.)	(in.)	(ksi)	(MPa)	(in.)	(in.)	(in.)	(ksi)	(MPa)
1/4	0.375	0.030	0.89	6.14	2	2.125	0.070	0.37	2.55
3/8	0.500	0.035	0.79	5.45	2-1/2	2.625	0.080	0.34	2.34
1/2	0.625	0.040	0.67	4.62	3	3.125	0.090	0.32	2.21
5/8	0.750	0.042	0.64	4.41	3-1/2	3.625	0.100	0.31	2.14
3/4	0.875	0.045	0.58	4.00	4	4.125	0.110	0.30	2.07
1	1.125	0.050	0.50	3.45	5	5.125	0.125	0.27	1.86
1-1/4	1.375	0.055	0.45	3.10	6	6.125	0.140	0.25	1.72
1-1/2	1.625	0.060	0.42	2.90	8	8.125	0.200	0.28	1.93

* Annealed properties assumed.

Copper Tubing

- Seamless, hard drawn, 20-ft straight lengths.
- Copper, UNS C12200, Type M, plumbing use.
- Per ASTM B88.
- Allowable stress: SE = 6000 psi.*

Table D-18. Copper, Type M.

Water tube size	o.d	Nominal wall thickness	MAWP*		Water tube size	o.d	Nominal wall thickness	MAWP*	
(in.)	(in.)	(in.)	(ksi)	(MPa)	(in.)	(in.)	(in.)	(ksi)	(MPa)
3/8	0.500	0.025	0.56	3.86	2-1/2	2.625	0.065	0.27	1.86
1/2	0.625	0.028	0.50	3.45	3	3.125	0.072	0.25	1.72
3/4	0.875	0.032	0.40	2.76	3-1/2	3.625	0.083	0.25	1.72
1	1.125	0.035	0.34	2.34	4	4.125	0.095	0.25	1.72
1-1/4	1.375	0.042	0.34	2.34	5	5.125	0.109	0.23	1.59
1-1/2	1.625	0.049	0.34	2.34	6	6.125	0.122	0.22	1.52
2	2.125	0.058	0.30	2.07	8	8.125	0.170	0.23	1.59

* Annealed properties assumed.

Carbon Steel Tubing

- Seamless, cold drawn, stress relieved.
- Carbon steel, AISI MT-1016/1018.
- Per ASTM A519.
- Allowable stress: SE = 15000 psi.

Table D-19. Carbon steel.

o.d	Nominal wall thickness	MAWP*		o.d	Nominal wall thickness	MAWP*	
(in.)	(in.)	(ksi)	(MPa)	(in.)	(in.)	(ksi)	(MPa)
0.250	0.065	8.93	61.60	2.500	0.120	1.40	9.66
0.500	0.035	2.02	13.93	2.500	0.3125	3.88	26.76
0.500	0.049	2.88	19.86	2.500	0.375	4.76	32.83
0.500	0.065	3.93	27.10	2.750	0.120	1.27	8.76
0.625	0.065	3.08	21.24	3.000	0.120	1.16	8.00
0.750	0.035	1.32	9.10	3.000	0.313	3.18	21.93
0.750	0.065	2.53	17.45	3.000	0.438	4.61	31.79
0.750	0.1875	8.38	57.79	3.125	0.120	1.11	7.66
0.875	0.065	2.15	14.83	3.250	0.065	0.57	3.93
0.875	0.120	4.17	28.76	3.750	0.250	1.97	13.59
1.000	0.035	0.98	6.76	3.750	0.375	3.04	20.97
1.000	0.065	1.87	12.90	3.750	0.625	5.36	36.97
1.000	0.1875	5.96	41.10	4.000	0.065	4.90	33.80
1.125	0.065	1.65	11.38	4.000	0.120	0.86	5.93
1.250	0.065	1.48	10.21	4.000	0.188	1.315	9.48
1.375	0.065	1.34	9.24	4.000	0.250	1.84	12.69
1.375	0.120	2.55	17.59	4.000	0.500	3.88	26.76
1.500	0.065	1.22	8.41	4.250	0.250	1.73	11.93
1.500	0.120	2.33	16.07	4.750	0.120	0.72	4.97
1.500	0.1875	3.77	26.00	5.000	0.250	1.46	10.07
1.625	0.065	1.12	7.72	5.250	0.250	1.39	9.59
1.750	0.250	4.37	30.14	5.500	0.120	0.62	4.28
2.000	0.065	0.93	6.41	5.500	0.250	1.32	9.10
2.000	0.375	6.04	41.06	6.000	0.188	0.90	6.21
2.250	0.120	1.56	10.76	6.000	0.375	1.84	12.69
2.250	0.250	3.41	23.52	6.750	0.250	1.07	7.38
2.375	0.4375	5.92	40.81	8.000	0.375	1.37	9.45
2.500	0.049	0.55	3.79	10.000	0.250	0.71	4.90

* Based on SAE J 1397, MT-1016.

Stainless Steel Tubing

- Seamless, annealed, cold drawn, pickled.
- Type TP 304, UNS S30400.
- Per ASTM A213.
- Allowable stress: SE = 18800 psi.

Table D-20. Stainless steel.

o.d	Nominal wall thickness	MAWP		o.d	Nominal wall thickness	MAWP	
(in.)	(in.)	(ksi)	(MPa)	(in.)	(in.)	(ksi)	(MPa)
0.125	0.016	4.63	31.89	1.000	0.065	2.24	15.45
0.188	0.016	2.98	20.53	1.250	0.049	1.33	9.14
0.250	0.020	2.79	19.22	1.250	0.065	1.78	12.24
0.250	0.035	5.11	35.21	1.500	0.049	1.10	7.58
0.250	0.049	7.47	51.53	1.500	0.065	1.47	10.14
0.250	0.065	10.46	72.10	1.500	0.083	1.89	13.06
0.313	0.020	2.20	15.20	1.500	0.095	2.18	15.03
0.313	0.035	4.00	27.57	1.625	0.065	1.35	9.34
0.313	0.049	5.31	36.62	1.750	0.065	1.23	8.65
0.375	0.020	1.82	12.57	2.000	0.035	0.58	4.02
0.375	0.035	3.29	22.65	2.000	0.065	1.09	7.54
0.375	0.049	4.73	32.63	2.000	0.109	1.86	12.85
0.375	0.065	6.49	44.75	2.000	0.120	2.06	14.21
0.500	0.035	2.42	16.70	2.500	0.035	0.47	3.21
0.500	0.049	3.46	23.87	2.500	0.065	0.87	6.01
0.500	0.065	4.71	32.44	2.500	0.120	1.63	11.27
0.500	0.109	8.46	58.36	3.000	0.065	0.72	4.99
0.625	0.065	3.69	25.44	3.000	0.120	1.35	9.34
0.750	0.035	1.59	10.94	4.000	0.065	0.54	3.73
0.750	0.049	2.25	15.53	4.000	0.083	0.69	4.78
0.750	0.065	3.04	20.93	6.000	0.083	0.46	3.17
1.000	0.049	1.67	11.51				

Appendix E

Metric Guide

The alphabetical list of units in this appendix was extracted from the *Metric Practice Guide*, E 380-74, published by the American Society for Testing and Materials (ASTM).

An asterisk (*) in the "Multiply by" column indicates that the conversion factor is exact and that all subsequent digits are zero. Because the footnotes are typically historical and are not generally important to LLNL engineers, they have not been included. They are available in the ASTM *Metric Practice Guide* for those interested.

For those unfamiliar with computer exponent notation, following is an example of how to use this list to convert a number expressed in an English unit to the corresponding number in System International (SI) units.

> Suppose you want to express a bar of pressure in SI units. The list heading says: "To convert from bar to pascal (Pa), multiply by 1.000 000*E+05," or by 1×10^5. Therefore, 1 bar = 1×10^5 Pa = 100 000 Pa, which is also 100 kPa, or 0.1 MPa.
>
> The basic SI pressure unit is the pascal (Pa), which is equal to 0.000145 psig (1 psig = 6895 Pa). One kilopascal (kPa) is equal to 0.145 psig. Notice that 100 kPa absolute = 14.5 psia = 1 bar, which is approximately equal to 1 atmosphere of pressure. One megapascal (MPa), which is 10^6 Pa, is equal to 145 psig.

ALPHABETICAL LIST OF UNITS

(Symbols of SI units given in parentheses)

To convert from	to	Multiply by
abampere	ampere (A)	1.000 000* E+01
abcoulomb	coulomb (C)	1.000 000* E+01
abfarad	farad (F)	1.000 000* E+09
abhenry	henry (H)	1.000 000* E–09
abmho	siemens (S)	1.000 000* E+09
abohm	ohm (Ω)	1.000 000* E–09
abvolt	volt (V)	1.000 000* E–08
acre foot (U.S. survey)	meter^3 (m^3)	1.233 489 E+03
acre (U.S. survey)	meter^2 (m^2)	4.046 873 E+03
ampere hour	coulomb (C)	3.600 000* E+03
are	meter^2 (m^2)	1.000 000* E+02
angstrom	meter (m)	1.000 000* E–10
astronomical unit	meter (m)	1.495 979 E+11
atmosphere (standard)	pascal (Pa)	1.013 250* E+05
atmosphere (technical = 1 kgf/cm^2)	pascal (Pa)	9.806 650* E+04
bar	pascal (Pa)	1.000 000* E+05
barn	meter^2 (m^2)	1.000 000* E–28
barrel (for petroleum, 42 gal)	meter^3 (m^3)	1.589 873 E–01
board foot	meter^3 (m^3)	2.359 737 E–03
British thermal unit (International Table)	joule (J)	1.055 056 E+03
British thermal unit (mean)	joule (J)	1.055 87 E+03
British thermal unit (thermochemical)	joule (J)	1.054 350 E+03
British thermal unit (39°F)	joule (J)	1.059 67 E+03
British thermal unit (59°F)	joule (J)	1.054 80 E+03

To convert from	to	Multiply by
British thermal unit (60°F)	joule (J)	1.054 68 E+03
Btu (International Table) ·ft/h ·ft^2 ·°F (*k*, thermal conductivity)	watt per meter kelvin (W/m ·K)	1.730 735 E+00
Btu (thermochemical) ·ft/h ·ft^2 ·°F (*k*, thermal conductivity)	watt per meter kelvin (W/m ·K)	1.729 577 E+00
Btu (International Table) ·in/h ·ft^2 ·°F (*k*, thermal conductivity)	watt per meter kelvin (W/m ·K)	1.442 279 E–01
Btu (thermochemical) ·in/h ·ft^2 ·°F (*k*, thermal conductivity)	watt per meter kelvin (W/m ·K)	1.441 314 E–01
Btu (International Table) ·in/s ·ft^2 ·°F (*k*, thermal conductivity)	watt per meter kelvin (W/m ·K)	5.192 204 E+02
Btu (thermochemical) ·in/s ·ft^2 ·°F (*k*, thermal conductivity)	watt per meter kelvin (W/m ·K)	5.188 732 E+02
Btu (International Table)/h	watt (W)	2.930 711 E–01
Btu (International Table)/s	watt (W)	1.055 056 E+03
Btu (thermochemical)/h	watt (W)	2.928 751 E–01
Btu (thermochemical)/min	watt (W)	1.757 250 E+01
Btu (thermochemical)/s	watt (W)	1.054 350 E+03
Btu (International Table)/ft^2	joule per meter^2 (J/m^2)	1.135 653 E+04
Btu (thermochemical)/ft^2	joule per meter^2 (J/m^2)	1.134 893 E+04
Btu (thermochemical)/ft^2 ·h	watt per meter^2 (W/m^2)	3.152 481 E+00
Btu (thermochemical)/ft^2 ·min	watt per meter^2 (W/m^2)	1.891 489 E+02
Btu (thermochemical)/ft^2 ·s	watt per meter^2 (W/m^2)	1.134 893 E+04
Btu (thermochemical)/in^2 ·s	watt per meter^2 (W/m^2)	1.634 246 E+06
Btu (International Table)/h ·ft^2 ·°F (*C*, thermal conductance)	watt per meter^2 kelvin (W/m^2 ·K)	5.678 263 E+00
Btu (thermochemical)/h ·ft^2 ·°F (*C*, thermal conductance)	watt per meter^2 kelvin (W/m^2 ·K)	5.674 466 E+00

To convert from	to	Multiply by
Btu (International Table)/s ·ft^2 ·°F	watt per meter^2 kelvin (W/m^2 ·K)	2.044 175 E+04
Btu (thermochemical)/s ·ft^2 ·°F	watt per meter^2 kelvin (W/m^2 ·K)	2.042 808 E+04
Btu (International Table)/lb	joule per kilogram (J/kg)	2.326 000* E+03
Btu (thermochemical)/lb	joule per kilogram (J/kg)	2.324 444 E+03
Btu (International Table)/lb ·°F (*c*, heat capacity	joule per kilogram kelvin (J/kg ·K)	4.186 800* E+03
Btu (thermochemical)/lb ·°F (*c*, heat capacity	joule per kilogram kelvin (J/kg ·K)	4.184 000* E+03
bushel (U.S)	meter^3 (m^3)	3.523 907 E–02
caliber (inch)	meter (m)	2.540 000* E–02
calorie (International Table)	joule (J)	4.186 800* E+00
calorie (mean)	joule (J)	4.190 02 E+00
calorie (thermochemical)	joule (J)	4.184 000* E+00
calorie (15°C)	joule (J)	4.185 80 E+00
calorie (20°C)	joule (J)	4.181 90 E+00
calorie (kilogram, International Table)	joule (J)	4.186 800* E+03
calorie (kilogram, mean)	joule (J)	4.190 02 E+03
calorie (kilogram, thermochemical)	joule (J)	4.184 000* E+03
cal (thermochemical)/cm^2	joule per meter^2 (J/m^2)	4.184 000* E+04
cal (International Table)/g	joule per kilogram (J/kg)	4.186 800* E+03
cal (thermochemical)/g	joule per kilogram (J/kg)	4.184 000* E+03
cal (International Table)/g ·°C	joule per kilogram kelvin (J/kg ·K)	4.186 800* E+03
cal (thermochemical)/g ·°C	joule per kilogram kelvin (J/kg ·K)	4.184 000* E+03
cal (thermochemical)/min	watt (W)	6.973 333 E–02
cal (thermochemical)/s	watt (W)	4.184 000* E+00

To convert from	to	Multiply by
cal (thermochemical)/$cm^2 \cdot$min	watt per meter^2 (W/m^2)	6.973 333 E+02
cal (thermochemical)/$cm^2 \cdot$s	watt per meter^2 (W/m^2)	4.184 000* E+04
cal (thermochemical)/$cm^2 \cdot$s $\cdot$°C	watt per meter^2 kelvin (W/m $\cdot$K)	4.184 000* E+02
carat (metric)	kilogram (kg)	2.000 000* E+04
centimeter of mercury (0°C)	pascal (Pa)	1.333 22 E+03
centimeter of water (4°C)	pascal (Pa)	9.806 38 E+01
centipoise	pascal second (Pa $\cdot$s)	1.000 000* E–03
centistokes	meter^2 per second (m^2/s)	1.000 000* E–06
circular mil	meter^2 (m^2)	5.067 075 E–10
clo	kelvin meter^2 per watt (K $\cdot m^2$/W)	2.003 712 E–01
cup	meter^3 (m^3)	2.365 882 E–04
curie	becquerel (Bq)	3.700 000* E+10
day (mean solar)	second (s)	8.640 000 E+04
day (sidereal)	second (s)	8.616 409 E+04
degree (angle)	radian (rad)	1.745 329 E–02
degree Celsius	kelvin (K)	$t_K = t_{°C} + 273.15$
degree centigrade	kelvin (K)	$t_K = t_{°C} + 273.15$
degree Fahrenheit	degree Celsius	$t_{°C} = (t_{°F} - 32)/1.8$
degree Fahrenheit	kelvin (K)	$t_K = (t_{°F} + 459.67)/1.8$
degree Rankine	kelvin (K)	$t_K = t_{°R}/1.8$
°F $\cdot$h $\cdot ft^2$/Btu (International Table) (*R*, thermal resistance)	kelvin meter^2 per watt (K $\cdot m^2$/W)	1.761 102 E–01
°F $\cdot$h $\cdot ft^2$/Btu (thermochemical) (*R*, thermal resistance)	kelvin meter^2 per watt (K $\cdot m^2$/W)	1.761 280 E–01
denier	kilogram per meter (kg/m)	1.111 111 E–07
dyne	newton (N)	1.000 000* E–05
dyne $\cdot$cm	newton meter (N $\cdot$m)	1.000 000* E–07

To convert from	to	Multiply by
dyne/cm^2	pascal (Pa)	1.000 000* E–01
electronvolt	joule (J)	1.602 19 E–19
EMU of capacitance	farad (F)	1.000 000* E+09
EMU of current	ampere (A)	1.000 000* E+01
EMU of electric potential	volt (V)	1.000 000* E–08
EMU of inductance	henry (H)	1.000 000* E–09
EMU of resistance	ohm (Ω)	1.000 000* E–09
ESU of capacitance	farad (F)	1.112 650 E–12
ESU of current	ampere (A)	3.335 6 E–10
ESU of electric potential	volt (V)	2.997 9 E+02
ESU of inductance	henry (H)	8.987 554 E+11
ESU of resistance	ohm (Ω)	8.987 554 E+11
erg	joule (J)	1.000 000* E–07
erg/(cm^2·s)	watt per meter^2 (W/m^2)	1.000 000* E–03
erg/s	watt (W)	1.000 000* E–07
faraday (based on carbon-12)	coulomb (C)	9.648 70 E+04
faraday (chemical)	coulomb (C)	9.649 57 E+04
faraday (physical)	coulomb (C)	9.652 19 E+04
fathom	meter (m)	1.828 8 E+00
fermi (femtometer)	meter (m)	1.000 000* E–15
fluid ounce (U.S.)	meter^3 (m^3)	2.957 353 E–05
foot	meter (m)	3.048 000* E–01
foot (U.S. survey)	meter (m)	3.048 006 E–01
foot of water (39.2°F)	pascal (Pa)	2.988 98 E+03
ft^2	meter^2 (m^2)	9.290 304* E–02
ft^2/h (thermal diffusivity)	meter^2 per second (m^2/s)	2.580 640* E–05
ft^2/s	meter^2 per second (m^2/s)	9.290 304* E–02
ft^3 (volume; section modulus)	meter^3 (m^3)	2.831 685 E–02
ft^3/min	meter^3 per second (m^3/s)	4.719 474 E–04

To convert from	to	Multiply by
ft^3/s	meter^3 per second (m^3/s)	2.831 685 E–02
ft^4 (moment of section)	meter^4 (m^4)	8.630 975 E–03
ft/h	meter per second (m/s)	8.466 667 E–05
ft/min	meter per second (m/s)	5.080 000* E–03
ft/s	meter per second (m/s)	3.048 000* E–01
ft/s^2	meter per second^2 (m/s^2)	3.048 000* E–01
footcandle	lux (lx)	1.076 391 E+01
footlambert	candela per meter^2 (cd/m^2)	3.426 259 E+00
ft·lbf	joule (J)	1.355 818 E+00
ft·lbf/h	watt (W)	3.766 161 E–04
ft·lbf/min	watt (W)	2.259 697 E–02
ft·lbf/s	watt (W)	1.355 818 E+00
ft·poundal	joule (J)	4.214 011 E–02
free fall, standard (g)	meter per second^2 (m/s^2)	9.806 650* E+00
gal	meter per second^2 (m/s^2)	1.000 000* E–02
gallon (Canadian liquid)	meter^3 (m^3)	4.546 090 E–03
gallon (U.K. liquid)	meter^3 (m^3)	4.546 092 E–03
gallon (U.S. dry)	meter^3 (m^3)	4.404 884 E–03
gallon (U.S. liquid)	meter^3 (m^3)	3.785 412 E–03
gallon (U.S. liquid) per day	meter^3 per second (m^3/s)	4.381 264 E–08
gallon (U.S. liquid) per minute	meter^3 per second (m^3/s)	6.309 020 E–05
gallon (U.S. liquid) per per hp·h (SFC, specific fuel consumption)	meter^3 per joule (m^3/J)	1.410 089 E–09
gamma	tesla (T)	1.000 000* E–09
gauss	tesla (T)	1.000 000* E–04
gilbert	ampere (A)	7.957 747 E–01
gill (U.K.)	meter^3 (m^3)	1.420 654 E–04
gill (U.S.)	meter^3 (m^3)	1.182 941 E–04

To convert from	to	Multiply by
grad	degree (angular)	9.000 000* E–01
grad	radian (rad)	1.570 796 E–02
grain (1/7000 lb avoirdupois)	kilogram (kg)	6.479 891* E–05
grain (lb avoirdupois/7000) gal (U.S. liquid)	kilogram per meter^3 (kg/m^3)	1.711 806 E–02
gram	kilogram (kg)	1.000 000* E–03
g/cm^3	kilogram per meter^3 (kg/m^3)	1.000 000* E+03
gram-force/cm^2	pascal (Pa)	9.806 650 E+01
hectare	meter^2 (m^2)	1.000 000* E+04
horsepower (550 ft ·lbf/s)	watt (W)	7.456 999 E+02
horsepower (boiler)	watt (W)	9.809 50 E+03
horsepower (electric)	watt (W)	7.460 000* E+02
horsepower (metric)	watt (W)	7.354 99 E+02
horsepower (water)	watt (W)	7.460 43 E+02
horsepower (U.K.)	watt (W)	7.457 0 E+02
hour (mean solar)	second (s)	3.600 000 E+03
hour (sidereal)	second (s)	3.590 170 E+03
hundredweight (long)	kilogram (kg)	5.080 235 E+01
hundredweight (short)	kilogram (kg)	4.535 924 E+01
inch	meter (m)	2.540 000* E–02
inch of mercury (32°F)	pascal (Pa)	3.386 38 E+03
inch of mercury (60°F)	pascal (Pa)	3.376 85 E+03
inch of water (39.2°F)	pascal (Pa)	2.490 82 E+02
inch of water (60°F)	pascal (Pa)	2.488 4 E+02
in^2	meter^2 (m^2)	6.451 600* E–04
in^3 (volume; section modulus)	meter^3 (m^3)	1.638 706 E–05
in^3/min	meter^3 per second (m^3/s)	2.731 177 E–07
in^4 (moment of section)	meter^4 (m^4)	4.162 314 E–07

To convert from	to	Multiply by
in/s	meter per second (m/s)	2.540 000* E–02
in/s^2	meter per $second^2$ (m/s^2)	2.540 000* E–02
kayser	1 per meter (1/m)	1.000 000* E+02
kelvin	degree Celsius	$t_{°C} = t_K - 273.15$
kilocalorie (International Table)	joule (J)	4.186 800* E+03
kilocalorie (mean)	joule (J)	4.190 02 E+03
kilocalorie (thermochemical)	joule (J)	4.184 000* E+03
kilocalorie (thermochemical)/min	watt (W)	6.973 333 E+01
kilocalorie (thermochemical)/s	watt (W)	4.184 000* E+03
kilogram-force (kgf)	newton (N)	9.806 650* E+00
kgf ·m	newton meter (N ·m)	9.806 650* E+00
kgf ·s^2/m (mass)	kilogram (kg)	9.806 650* E+00
kgf/cm^2	pascal (Pa)	9.806 650* E+04
kgf/m^2	pascal (Pa)	9.806 650* E+00
kgf/mm^2	pascal (Pa)	9.806 650* E+06
km/h	meter per second (m/s)	2.777 778 E–01
kilopond	newton (N)	9.806 650* E+00
kW ·h	joule (J)	3.600 000* E+06
kip (1000 lbf)	newton (N)	4.448 222 E+03
kip/in^2 (ksi)	pascal (Pa)	6.894 757 E+06
knot (international)	meter per second (m/s)	5.144 444 E–01
lambert	candela per $meter^2$ (cd/m^2)	$1/\pi$ * E+04
lambert	candela per $meter^2$ (cd/m^2)	3.183 099 E+03
langley	joule per $meter^2$ (J/m^2)	4.184 000* E+04
league	meter (m)	
light year	meter (m)	9.460 55 E+15
liter	$meter^3$ (m^3)	1.000 000* E–03
maxwell	weber (Wb)	1.000 000* E–08

To convert from	to	Multiply by
mho	siemens (S)	1.000 000* E+00
microinch	meter (m)	2.540 000* E–08
micron	meter (m)	1.000 000* E–06
mil	meter (m)	2.540 000* E–05
mile (international)	meter (m)	1.609 344* E+03
mile (statute)	meter (m)	1.609 3 E+03
mile (U.S. survey)	meter (m)	1.609 347 E+03
mile (international nautical)	meter (m)	1.852 000* E+03
mile (U.K. nautical)	meter (m)	1.853 184* E+03
mile (U.S. nautical)	meter (m)	1.852 000* E+03
mi^2 (international)	$meter^2$ (m^2)	2.589 988 E+06
mi^2 (U.S. survey)	$meter^2$ (m^2)	2.589 998 E+06
mi/h (international)	meter per second (m/s)	4.470 400* E–01
mi/h (international)	kilometer per hour (km/h)	1.609 344* E+00
mi/min (international)	meter per second (m/s)	2.682 240* E+01
mi/s (international)	meter per second (m/s)	1.609 344* E+03
millibar	pascal (Pa)	1.000 000* E+02
millimeter of mercury (0°C)	pascal (Pa)	1.333 22 E+02
minute (angle)	radian (rad)	2.908 882 E–04
minute (mean solar)	second (s)	6.000 000 E+01
minute (sidereal)	second (s)	5.983 617 E+01
month (mean calendar)	second (s)	2.628 000 E+06
oersted	ampere per meter (A/m)	7.957 747 E+01
ohm centimeter	ohmmeter (Ω·m)	1.000 000* E–02
ohm circular-mil per foot	ohm $millimeter^2$ per meter (Ω·mm^2/m)	1.662 426 E–03
ounce (avoirdupois)	kilogram (kg)	2.834 952 E–02
ounce (troy or apothecary)	kilogram (kg)	3.110 348 E–02
ounce (U.K. fluid)	$meter^3$ (m^3)	2.841 307 E–05

To convert from	to	Multiply by
ounce (U.S. fluid)	meter^3 (m^3)	2.957 353 E–05
ounce-force	newton (N)	2.780 139 E–01
ozf·in	newton meter (N·m)	7.061 552 E–03
oz (avoirdupois)/gal (U.K. liquid)	kilogram per meter^3 (kg/m^3)	6.236 021 E+00
oz (avoirdupois)/gal (U.S. liquid)	kilogram per meter^3 (kg/m^3)	7.489 152 E+00
oz (avoirdupois)/in^3	kilogram per meter^3 (kg/m^3)	1.729 994 E+03
oz (avoirdupois)/ft^2	kilogram per meter^2 (kg/m^2)	3.051 517 E–01
oz (avoirdupois)/yd^2	kilogram per meter^2 (kg/m^2)	3.390 575 E–02
parsec	meter (m)	3.085 678 E+16
peck (U.S.)	meter^3 (m^3)	8.809 768 E–03
pennyweight	kilogram (kg)	1.555 174 E–03
perm (0°C)	kilogram per pascal second meter^2 (kg/Pa·s·m^2)	5.721 35 E–11
perm (23°C)	kilogram per pascal second meter^2 (kg/Pa·s·m^2)	5.745 25 E–11
perm·in (0°C)	kilogram per pascal second meter (kg/Pa·s·m)	1.453 22 E–12
perm·in (23°C)	kilogram per pascal second meter (kg/Pa·s·m)	1.459 29 E–12
phot	lumen per meter^2 (lm/m^2)	1.000 000* E+04
pica (printer's)	meter (m)	4.217 518 E–03
pint (U.S. dry)	meter^3 (m^3)	5.506 105 E–04
pint (U.S. liquid)	meter^3 (m^3)	4.731 765 E–04

To convert from	to	Multiply by
point (printer's)	meter (m)	3.514 598* E–04
poise (absolute viscosity)	pascal second (Pa ·s)	1.000 000* E–01
pound (lb avoirdupois)	kilogram (kg)	4.535 924 E–01
pound (troy or apothecary)	kilogram (kg)	3.732 417 E–01
lb ·ft^2 (moment of inertia)	kilogram meter^2 (kg ·m^2)	4.214 011 E–02
lb ·in^2 (moment of inertia)	kilogram meter^2 (kg ·m^2)	2.926 397 E–04
lb/ft ·h	pascal second (Pa ·s)	4.133 789 E–04
lb/ft ·s	pascal second (Pa ·s)	1.488 164 E+00
lb/ft^2	kilogram per meter^2 (kg/m^2)	4.882 428 E+00
lb/ft^3	kilogram per meter^3 (kg/m^3)	1.601 846 E+01
lb/gal (U.K. liquid)	kilogram per meter^3 (kg/m^3)	9.977 633 E+01
lb/gal (U.S. liquid)	kilogram per meter^3 (kg/m^3)	1.198 264 E+02
lb/h	kilogram per second (kg/s)	1.259 979 E–04
lb/hp ·h (SFC, specific fuel consumption)	kilogram per joule (kg/J)	1.689 659 E–07
lb/in^3	kilogram per meter^3 (kg/m^3)	2.767 990 E+04
lb/min	kilogram per second (kg/s)	7.559 873 E–03
lb/s	kilogram per second (kg/s)	4.535 924 E–01
lb/yd^3	kilogram per meter^3 (kg/m^3)	5.932 764 E–01
poundal	newton (N)	1.382 550 E–01
poundal/ft^2	pascal (Pa)	1.488 164 E+00
poundal ·s/ft^2	pascal second (Pa ·s)	1.488 164 E+00
pound ·force (lbf)	newton (N)	4.448 222 E+00
lbf ·ft	newton meter (N ·m)	1.355 818 E+00

To convert from	to	Multiply by
lbf·ft/in	newton meter per meter (N·m/m)	5.337 866 E+01
lbf·in	newton meter (N·m)	1.129 848 E–01
lbf·in/in	newton meter per meter (N·m/m)	4.448 222 E+00
lbf·s/ft^2	pascal second (Pa·s)	4.788 026 E+01
lbf·s/in^2	pascal second (Pa·s)	6.894 757 E+03
lbf/ft	newton per meter (N/m)	1.459 390 E+01
lbf/ft^2	pascal (Pa)	4.788 026 E+01
lbf/in	newton per meter (N/m)	1.751 268 E+02
lbf/in^2 (psig)	pascal (Pa)	6.894 757 E+03
lbf/lb [thrust/weight (mass) ratio]	newton per kilogram (N/kg)	9.806 650 E+00
quart (U.S. dry)	meter^3 (m^3)	1.101 221 E–03
quart (U.S. liquid)	meter^3 (m^3)	9.463 529 E–04
rad (radiation dose absorbed)	gray (Gy)	1.000 000* E–02
rhe	1 per pascal second (1/Pa·s)	1.000 000* E+01
rod	meter (m)	
roentgen	coulomb per kilogram (C/kg)	2.58 E–04
second (angle)	radian (rad)	4.848 137 E–06
second (sidereal)	second (s)	9.972 696 E–01
section	meter^2 (m^2)	
shake	second (s)	1.000 000* E–08
slug	kilogram (kg)	1.459 390 E+01
slug/ft·s	pascal second (Pa·s)	4.788 026 E+01
slug/ft^3	kilogram per meter^3 (kg/m^3)	5.153 788 E+02
statampere	ampere (A)	3.335 640 E–10
statcoulomb	coulomb (C)	3.335 640 E–10

To convert from	to	Multiply by
statfarad	farad (F)	1.112 650 E–12
stathenry	henry (H)	8.987 554 E–11
statmho	siemens (S)	1.112 650 E+12
statohm	ohm (Ω)	8.987 554 E+11
statvolt	volt (V)	2.997 925 E+02
stere	meter^3 (m^3)	1.000 000* E+00
stilb	candela per meter^2 (cd/m^2)	1.000 000* E+04
stokes (kinematic viscosity)	meter^2 per second (m^2/s)	1.000 000* E–04
tablespoon	meter^3 (m^3)	1.478 676 E–05
teaspoon	meter^3 (m^3)	4.928 922 E–06
tex	kilogram per meter (kg/m)	1.000 000* E+06
therm	joule (J)	1.055 056 E+08
ton (assay)	kilogram (kg)	2.916 667 E–02
ton (long, 2240 lb)	kilogram (kg)	1.016 047 E+03
ton (metric)	kilogram (kg)	1.000 000* E+03
ton (nuclear equivalent of TNT)	joule (J)	4.184 E+09
ton (refrigeration)	watt (W)	3.516 800 E+03
ton (register)	meter^3 (m^3)	2.831 685 E+00
ton (short, 2000 lb)	kilogram (kg)	9.071 847 E+02
ton (long)/yd^3	kilogram per meter^3 (kg/m^3)	1.328 939 E+03
ton (short)/yd^3	kilogram per meter^3 (kg/m^3)	1.186 553 E+03
ton (short)/h	kilogram per second (kg/s)	2.519 958 E–01
ton-force (2000 lbf)	newton (N)	8.896 444 E+03
tonne	kilogram (kg)	1.000 000* E+03
torr (mm Hg, 0°C)	pascal (Pa)	1.333 22 E+02
township	meter^2 (m^2)	
unit pole	weber (Wb)	1.256 637 E–07

To convert from	to	Multiply by
W ·h	joule (J)	3.600 000* E+03
W ·s	joule (J)	1.000 000* E+00
W/cm^2	watt per $meter^2$ (W/m^2)	1.000 000* E+04
W/in^2	watt per $meter^2$ (W/m^2)	1.550 003 E+03
yard	meter (m)	9.144 000* E–01
yd^2	meter ^{2} (m^2)	8.361 274 E–01
yd^3	meter ^{3} (m^3)	7.645 549 E–01
yd^3/min	$meter^3$ per second (m^3/s)	1.274 258 E–02
year (365 days)	second (s)	3.153 600 E+07
year (sidereal)	second (s)	3.155 815 E+07
year (tropical)	second (s)	3.155 693 E+07

Appendix F

Joint Efficiencies

Table 2. Maximum allowable joint efficiencies for arc and gas-welded joints (reproduced from Table UW-12 in ASME Boiler and Pressure Vessel Code).

No.	Type of Joint Description	Limitation	Degree of Examination		
			(a) Fully Radio-graphed[1]	(b) Spot Examined[2]	(c) Not Spot Examined[3]
(1)	Butt joints as attained by double-welding or by other means which will obtain the same quality of deposited weld metal on the inside and outside weld surfaces to agree with the requirements of UW-35. Welds using metal backing strips which remain in place are excluded.	None	1.00	0.85	0.70
(2)	Single-welded butt joint with backing strip other than those included under (1)	(a) None except as in (b) below (b) Butt weld with one plate off-set—for circumferential joints only, see UW-13(c) and Fig. UW-13.1 sketch (k).	0.90	0.80	0.65
(3)	Single-welded butt joint without use of backing strip	Circumferential joints only, not over 5/8-in. thick and not over 24 in. outside diameter.	—	—	0.60
(4)	Double fill fillet lap joint	Longitudinal joints not over 3/8-in. thick. Circumferential joints not over 5/8-in. thick.	—	—	0.50
(5)	Single fill fillet lap joints with plug welds conforming to UW-17	(a) Circumferential joints[4] for attachment of heads not over 24 in. outside diameter to shells not over 1/2-in. thick. (b) Circumferential joints for the attachment to shells of jackets not over 5/8 in. in nominal thickness where the distance from the center of the plug weld to the edge of the plate is not less than 1 1/2 times the diameter of the hole for the plug.			

Table 2. Maximum allowable joint efficiencies for arc and gas-welded joints (reproduced from Table UW-12 in ASME Boiler and Pressure Vessel Code) (cont'd).

No.	Type of Joint Description	Limitation	Degree of Examination		
			(a) Fully Radio-graphed[1]	(b) Spot Examined[2]	(c) Not Spot Examined[3]
(6)	Single full fillet lap joints without plug welds	(a) For the attachment of heads convex to pressure to shells not over 5/8 in. required thickness, only with use of fillet weld on inside of shell; or (b) For attachment of heads having pressure on either side, to shells not over 24 in. inside diameter and not over 1/4 in. required thickness with fillet weld on outside of head flange only.	–	–	0.45

(1) See UW-12(a) and UW-51.

(2) See UW-12(b) and UW-52.

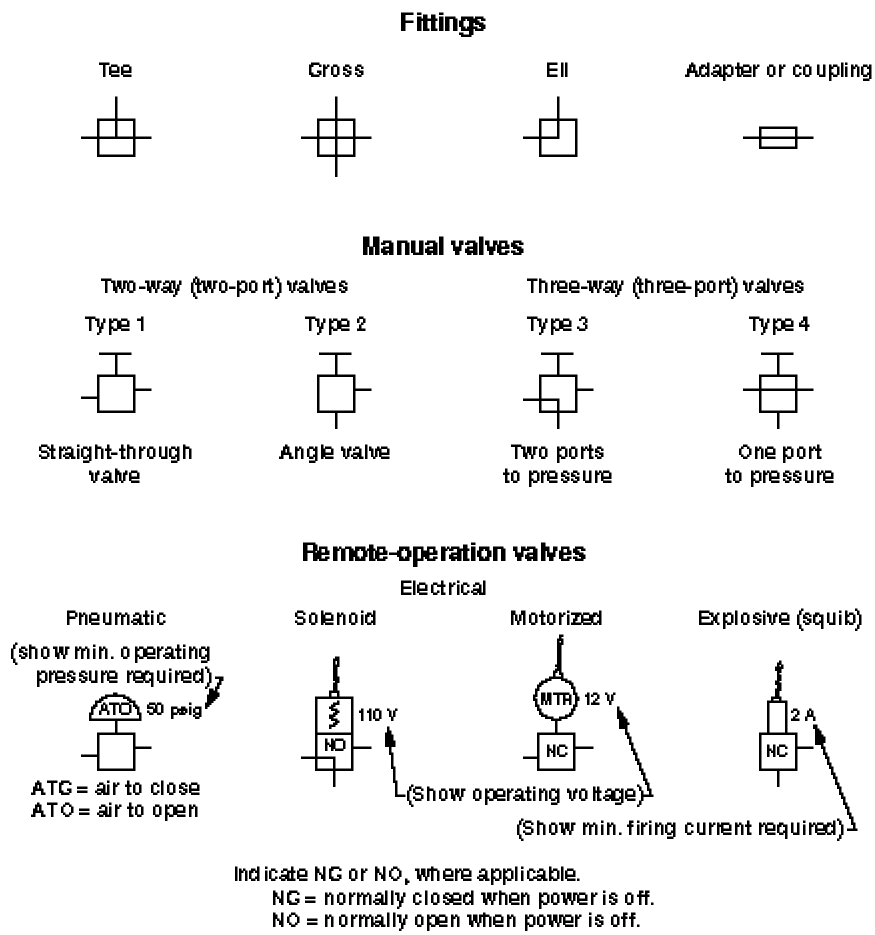
(3) The maximum allowable joint efficiencies shown in this column are the weld joint efficiencies multiplied by 0.80 (and rounded off to the nearest 0.05), to effect the basic reduction in allowable stress required by this division for welded vessels that are not spot examined. See UW-12(c).

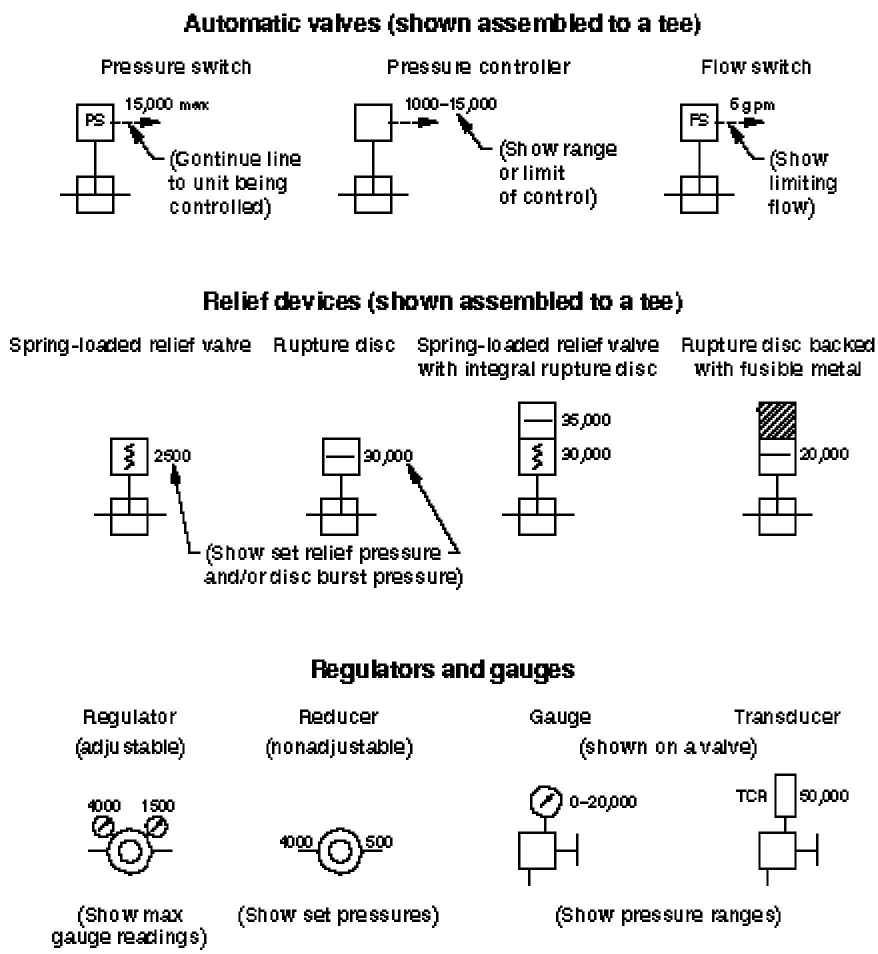
(4) Joints attaching hemispherical heads to shells are excluded.

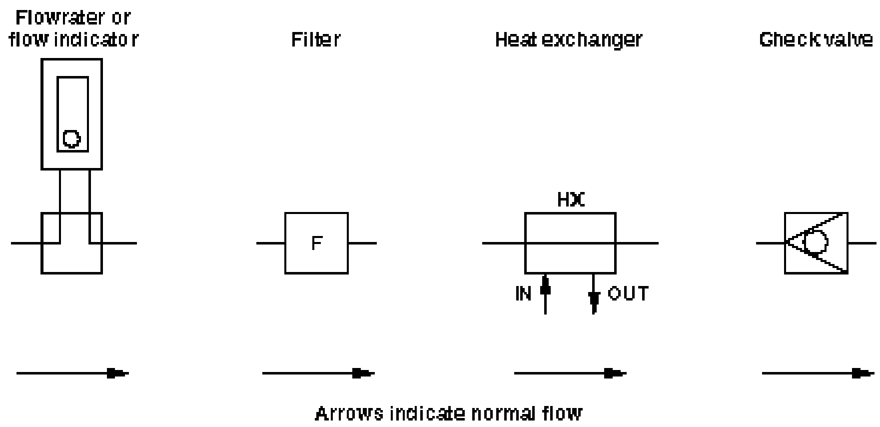
Appendix G

High-Pressure Drawing Symbols

The following high-pressure drawing symbols are standard in all LLNL engineering drawings.




Automatic valves (shown assembled to a tee)
Pressure switch
PS
15,000 max
(Continue line to unit being controlled)
Pressure controller
1000–15,000
(Show range or limit of control)
Flow switch
FS
6 gpm
(Show limiting flow)
Relief devices (shown assembled to a tee)
Spring-loaded relief valve
2500
Rupture disc
30,000
Spring-loaded relief valve with integral rupture disc
35,000
30,000
Rupture disc backed with fusible metal
20,000
(Show set relief pressure and/or disc burst pressure)
Regulators and gauges
Regulator (adjustable)
4000
1500
(Show max gauge readings)
Reducer (nonadjustable)
4000
500
(Show set pressures)
Gauge
0–20,000
Transducer
TCR
50,000
(shown on a valve)
(Show pressure ranges)


Flowrater or
flow indicator
Filter
Heat exchanger
Check valve
HX
F
IN
OUT
Arrows indicate normal flow